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NAVAL ENGINEERING PROGRESS.

A LECTURE BY ENGINEER-IN-CHIEF H. I. CONE, U. S. N., BEFORE THE U. S. NAVAL WAR COLLEGE, AUGUST 9, 1910.

MR. PRESIDENT AND GENTLEMEN:

In my paper today I am endeavoring to place before you in a brief and general way the developments of the past year in naval engineering, notably those affecting strategy and tactics.

SCOUT TRIALS.

Extensive trials have been carried on with the *Birmingham*, *Chester* and *Salem*—scout cruisers, similar in design except as regards their machinery installation. The *Birmingham* has reciprocating engines on two shafts; the *Chester*, Parsons turbines on four shafts, and the *Salem*, Curtis turbines on two shafts. The boilers on the *Birmingham* and *Salem* are identical in design, while those on the *Chester* are of similar type. (See Plate 4 on page 828, J. A. S. N. E., August, 1910.)

A brief of the economic results obtained in these trials is shown in the following table:

STEAM-CONSUMPTION TRIALS OF THE "BIRMINGHAM," "CHESTER" AND "SALEM."

.....	*Birmingham.*	*Chester.*			*Salem.*
Item.	2 engines.	6 turbines.	5 turbines.	4 turbines.	2 turbines.
Type of engines and number of turbines in use.	Four-cyl. triple-expansion vertical inverted engines.	Parsons turbine, with high-press. and intermed. pressure cruising turbines.	Parsons turbines, without high-pres. cruising turbine.	Parsons, two high-pressure and two low-pressure turbines.	Curtis turbines, on two shafts.
Number of shafts.....	2	4	4	4	2
Speed, knots.....	10.03	10.15	10.43	10.15	10.01
Revolutions per min..	74.3	206.3	212.2	206.8	139.3
I.H.P.....	1,050.0	.....	.....	.....	.....
Shaft horsepower.....	.....	908.0	828.5	746.6	719.6
Pounds of water per H.P. per hr. main engines only.....	20.94	28.1	40.95	44.9	42.0
Pounds of water per H.P. hour for all purposes.....	32.44	53.28	58.39	63.87	62.73
Total water per hour, main engines only..	21,988.0	25,492.0	33,927.0	33,539.0	30,182 0
Total water per hour, for all purposes.....	34,069.0	43,761.0	48,380.0	47,690.0	45,144.0
Speed, knots.....	24.15	18.03	22.62	24 67	25.12
Revolutions per min..	191.5	379.0	481.4	542.5	369.8
I.H.P.....	14,543.0	.....	.....	.....	.....
Shaft horsepower.....	.....	5,699.0	12,467.0	18,544.0	18,070.0
Pounds of water per H.P. per hr., main engines only.....	17.42	16.5	12.79	13.7	16.2
Pounds of water per H.P., for all purposes	20.44	20.26	15.07	15.96	18 89
Total water per hour, main engines only..	253,372.0	94,082.0	159,435.0	253,662.0	292,751.0
Total water per hour, for all purposes.....	297,300.0	115,458.0	187,933.0	296,054.0	341,486 0

At a speed of about ten knots the total water per hour for the main engines only is 21,988 pounds for the *Birmingham*, 25,492 for the *Chester* and 30,182 for the *Salem*.

On full power, with respective speeds of 24.15, 24.67 and 25.12 knots, this water consumption is 253,372 pounds for the *Birmingham*, 253,662 pounds for the *Chester* and 292,751

pounds for the *Salem*, or 17.42 pounds per I.H.P., 13.7 pounds per S.H.P. and 16.2 pounds per S.H.P., respectively.

The speeds on contract trials were: *Birmingham*, 24.325 knots; *Chester*, 26.52 knots; *Salem*, 25.947 knots.

An interesting point developed in the trials is the economy obtained in the Parsons system by the use of cruising turbines at low power. These cruising turbines have been omitted in recent British designs. At 10 knots, with the cruising turbines, a water rate of 28.1 pounds per S.H.P. was obtained, while without the cruising turbines this increased to 44.9 pounds per S.H.P.

In comparing the results of these trials it should be remembered that the Curtis turbines of the *Salem* represent a much earlier period of development than do the Parsons turbines of the *Chester*.

STEAM TRIALS OF THE "DELAWARE" AND "NORTH DAKOTA."

The trials of the *Delaware* and *North Dakota* present an interesting study in similar ships of the comparative efficiencies of reciprocating engines and Curtis turbines in battleships. These results are shown briefly in the following table:

COMPARISON OF TRIAL RESULTS OF THE "DELAWARE" WITH RECIPROCATING ENGINES, AND THE "NORTH DAKOTA" WITH CURTIS TURBINES.

.....	*Delaware.*	*North Dakota.*	*Delaware.*	*North Dakota.*	*Delaware.*	*North Dakota.*
Speed, knots	12.24	12.1	19.225	19.245	21.563	21.64
Water per indicated horsepower per hour, main engines only, pounds	15.32	20.6	12.99	12.7	13.42	13 8
Water per indicated horsepower per hour, for all purposes, pounds	21.05	23.94	14.52	15.92	14.8	14 408
Total water per hour for main engines only, pounds	55,000	76,000	205,000	238,000	318,000	349,000
Total water per hour for all purposes, pounds	83,463	90,984	250,549	266,711	422,931	450,965
Total water per knot, main engines only, pounds	4,494	6,281	10,663	12,367	14,469	16,128
Total water per knot, for all purposes, pounds	6,819	7,519	13,033	13,859	19,614	20,839

The *Delaware*, which is our latest and perhaps last reciprocating-engined battleship, showed such an excellent performance on her contract trial that there is doubt in the minds

of some of us as to the superiority of the turbine for a battleship of that type and speed. The steam consumption per I.H.P. of the main engines was 15.48 pounds at 12 knots, 12.7 pounds at 19 knots and 12.9 pounds at 21 knots.

As compared with the *North Dakota* the main engines of the *Delaware* showed a smaller steam consumption at all speeds, and at 12 knots this superior economy amounted to about one-third the total consumption. Translated into cruising radius, this superiority, if maintained on service, is significantly important.

The principal improvements in the *Delaware's* machinery over that of her predecessor's consist in:

(a) Improved cylinder ratios and reduced clearance in main engines;

(b) Forced lubrication of main bearings, crank pins, eccentrics and crosshead pins;

(c) The use of superheated steam.

Forced lubrication similar to that of the *Delaware* is being applied to the older vessels. During the present year it will be fitted to the *Vermont*, *New Hampshire*, *Georgia*, *North Carolina*, *California* and *Colorado*, and, if it proves successful, as rapidly as possible to the other armored vessels of the fleet.

Some difficulties have developed in connection with the use of superheated steam, due to erosion of valves in the steam lines of the *Delaware*, and there is still some doubt as to whether the economy gained is not counterbalanced by the increased difficulty of maintenance of plant. We are experimenting with metals suited to high temperatures, and hope to find a valve material to overcome these troubles.

THE BACKING POWER OF TURBINES AS COMPARED WITH RECIPROCATING ENGINES.

In the beginning of our experience with turbines our attention was naturally centered on their engineering characteristics, such as steam consumption and details of construc-

tion and operation. With the satisfactory development of these elements, we have been able to devote more attention to the tactical features involved as affecting the ease of handling of turbine vessels.

With reciprocating engines a backing power equal to the ahead power is afforded without any increase in weight except that of the backing eccentrics, rods and links. With turbines, however, the ability to reverse requires additional turbines, with a considerable increase in weight.

It is necessary, therefore, to restrict the backing power of turbines to that actually required by tactical considerations. The intention has been to provide a backing power estimated at 50 per cent. of the ahead power. In practice, however, the backing power of turbine vessels has been found not to exceed 40 per cent. of the ahead power.

Trials have recently been held with the three scout cruisers to determine the sufficiency of the backing power provided by Parsons and by Curtis turbines, and as compared with that obtained by reciprocating engines. The basis of comparison has been taken as the distance head reached by the vessel during the interval required to bring her dead in the water from a given speed ahead.

These trials were conducted with great care and thoroughness, the condition of displacement, sea, speed, and number of boilers in use, were practically identical for all, and the results may be taken as an authoritative indication of the relative backing powers of the three types of machinery as now installed.

The speeds selected for the trials were 10, 16, 22 and 24 knots. Three runs were made by each vessel at each of the two lower speeds, two runs at 22 knots, and one run at the high speed. At each speed only the boilers required for a sustained run at this speed were used.

There was no bottling up of steam preparatory to backing, and no speeding up of blowers during the backing interval. It was required that the boiler pressure be not reduced to such extent as to cause priming.

The results are shown in the following table:

COMPARATIVE BACKING DATA U. S. S. "BIRMINGHAM," "CHESTER" AND "SALEM."

Speed in knots.			Mean draught. feet and inches.			Displacement.			Time to stop from instant of signal for reversal.			Distance head reached in ship lengths.		
B.	C.	S.	B.	C.	S.	B.	C.	S.	B.	C.	S.	B.	C.	S.
10.00	9.85	10.00	19-07.25	19-08	19-08	4,855.5	4,950	4,877	1:10.3	1:52.7	1:20.4	1.5	2.2	1.8
16.07	15.79	16.00	19-07	19-07	19-07	4,851.0	4,842	4,846	1:21 2	2:06	1:27 3	2.6	4.3	3.0
21.75	22.09	22.00	19-06.5	19-04	19-07	4,835.5	4,785	4,846	1:03.0	2:08.5	1:37.0	3 0	5.4	4.2
23 40	24 25	24.00	19-06.375	19-04	19-07	4,832.5	4,785	4,846	1:09.0	2:31	1:40.5	3.2	6.5	4.7

It will be noted that at all speeds the reciprocating engine provides better backing power than the Curtis turbines, and that the latter is superior to the Parsons turbine.

The commanding officer of the *Chester* (with Parsons turbines) states: "However, under all the usual experiences of service, I have felt, during the past four months' experience, no uneasiness as to the adequacy of the backing power, provided I always had six boilers in operation." The *Chester* has twelve boilers, and on the trials six were used at 10 knots with natural draft.

The adequacy of the *Chester's* backing power is, I believe, debatable, and I submit it to you gentlemen, it being a tactical rather than an engineering question.

The average time required for reversing and the fall in boiler pressure during the backing interval were as follows:

	Average time to reverse, in seconds.	Average drop in boiler pressure, pounds.
Reciprocating engines.....	5.1	None.
Parsons turbines.....	8.2	82.4
Curtis turbines...	39.2	75.8

In the case of the *Birmingham* the average boiler pressure at the end of the backing interval was 1.5 pounds greater than at the beginning of the interval.

Similar trials are to be carried out with the *North Dakota* and *Delaware.*

The British battleship *Dreadnought*, displacing 17,900 tons, and with 23,000 S.H.P. in Parsons turbines, at a speed of 20 knots was stopped in three minutes in 5.9 ship lengths,

and at 12 knots in 4.2 ship lengths, the time for the latter speed not being given.

STRATEGICAL FEATURES GOVERNING OUR MACHINERY INSTALLATIONS.

The strategical features governing the design of our naval machinery are quite different in the case of other nations with whose vessels ours are ordinarily compared. Certain European nations, for instance, are building fleets for service in the North Sea. Their probable enemies are close at hand, the probable sphere of their naval activities is limited, and, as a result, their fleets must be prepared to make short changes of base at high speed. The result is apparent in recent British designs, in which the predominant features are high speed and simplicity of machinery at the expense of cruising radius. For their purposes the increased economy gained by the use of cruising turbines is not justified by the increased weight, space and complexity involved.

We, however, must build our ships for service beyond the seas. Economy of fuel at cruising speed is to us of paramount importance, and our problem becomes the difficult one of combining a maximum of cruising radius with an ability to make the speed required by tactical considerations.

This extends even to our destroyers. Their cruising radius should approximate that of the battleship fleet, and this at a sustained speed of 15 knots. We are now considering in the designs of the new destroyers methods of reducing their steam consumption at 15 knots, in an endeavor to build up their cruising radius at this speed. Two methods present themselves. In the one the cruising turbines hitherto shown in our three-shaft destroyers are replaced by small high-speed turbines connected by mechanical reduction gear to the low-pressure shafts. A clutch coupling is interposed to permit throwing out the cruising element at the higher speeds and when backing. The small turbine works the steam from the boiler pressure down to about atmospheric pressure, and exhausts through the main turbines, the power at 15 knots being about equally divided between the three shafts.

The other proposition is to install a small reciprocating engine in place of the high-speed turbine and reduction gear. On account of the low power required for 15 knots (about 450 H.P. on each shaft), and because of the comparatively high efficiency of the reciprocating engine when working in the range of pressures presented, the design of an engine which will be satisfactory on service is by no means difficult. A moderate piston speed and forced lubrication can be provided which should ensure ease of upkeep.

Either of these proposed cruising elements can be installed within the weight and space of the cruising turbines hitherto used, and each method promises an improvement in economy of about 20 per cent. The resulting increase in cruising radius may justify the departure from simplicity of plant entailed by the employment of one of these methods, particularly since total disability of the cruising elements would still leave a full-powered plant.

PROSPECTIVE DEVELOPMENTS IN METHODS OF PROPULSION.

Although there have been marked improvements in the design of recent turbine machinery, an all-turbine installation in a battleship is still unsatisfactory as regards economy of coal at cruising speeds, due to the inefficiency of the turbine at low peripheral speeds. There is, therefore, a well-defined need for a method of propulsion which, at cruising speeds, will allow a high-speed turbine to drive a slowly-revolving propeller shaft, thus conserving both turbine and propeller efficiencies. It is probable that three methods of propulsion, radically different from existing types and all aiming to accomplish this end, will be tried within the next two years.

HIGH-SPEED MARINE STEAM TURBINE WITH REDUCTION GEAR.

This form of drive is to be installed in the collier *Neptune*, building at the works of the Maryland Steel Company, Sparrows Point, Md. She is to be a twin-screw vessel, similar to the *Cyclops*, displacing 19,360 tons, with a speed of 14 knots.

Steam at a pressure of 200 pounds will be supplied by three double-end Scotch boilers to a Westinghouse-Parsons turbine for each shaft, each turbine developing about 4,000 S.H.P. on 1,500 revolutions at full power. Between each turbine and its propeller shaft is to be interposed a Melville-Macalpine gear, reducing the propeller speeds to 135 r.p.m. at 14 knots. Low speeds are obtained by slowing down the turbine, and astern turbines are provided for backing.

It is expected that, incidental to the trial of this method of propulsion, the installation in the collier will afford an opportunity to benefit by the broad experience of the Westinghouse Company in the development of details of the work. For instance, it is proposed to adapt the company's electro-pneumatic railway-switch-operating mechanism, to permit the operation of the machinery from the bridge. Again, the installation will permit a trial of the Leblanc air pump, as used by the company in its commercial work, and of radical changes in design and construction of the Parsons turbine.

This installation is being made on very liberal terms to the Government, the cost being that of the reciprocating engines originally contemplated, and the Government being duly guaranteed against possible failure.

The principal points to be developed with this installation are the extent of wear on the gears, quietness of operation, and the adaptability of the gears for reversing. In shop tests the gear has transmitted 4,500 S.H.P. for forty hours, with an efficiency in excess of 98 per cent., and with no apparent tendency to wear.

W. L. R. EMMET'S METHOD OF ELECTRIC PROPULSION.

Mr. W. L. R. Emmet, an engineer of the General Electric Company, has submitted a form of electric propulsion in which General Electric high-speed turbo-generators drive motors on the propeller shafts at speeds considered proper for propeller efficiency. He estimates an efficiency of transmission of 94 per cent. His installation in a battleship contemplates two turbo-generators, driving four motors on two main

shafts. By this multiplicity of units, combined with pole-changing devices on two of the motors, an economy close to the maximum is promised at all speeds above 10 knots.

In the case of a collier, whose cruising speed would probably be near her maximum, the installation proposed consists of two generators and two motors, one on each propeller shaft. The motors would not be arranged for pole changing, but would be equipped with resistance for reversing. At speeds below 11 knots one generator would be used on both motors. At a speed of 14 knots a water rate of 12.6 pounds per S.H.P. per hour is promised, and at 10 knots the water rate is stated to be less than 15 pounds.

It is hoped that this method of propulsion may be installed in a collier authorized by the last Congress, similar to the *Neptune*, under conditions similar to those governing the Westinghouse installation in the latter vessel.

Although many of the features of the electric drive proposed by Mr. Emmet have been developed as the result of actual installations in shore plants, there are so many novel features involved that the system will require careful scrutiny before its use in an important vessel is considered. It is difficult to predict what disadvantages, if any, will be found, but the following points will be the subject of special investigation in the trial installation:

(a) Weight as compared with all-turbine drive;

(b) Ventilation of the generators and motors, there being considerable heat generated in the armatures;

(c) Cooling of resistances. Under certain conditions there are large quantities of heat to be suddenly dissipated from the resistances.

(d) The possible failure of the machines or parts of the machines due to access of water;

(e) Complexity of accessories—switches, pole-changing devices, resistances, etc.;

(f) The degree of delicacy of speed control;

(g) The effect on chronometers, compasses and personnel of grounds, the tension being in excess of 2,000 volts.

THE MAVOR AND COULSON SYSTEM OF ELECTRIC DRIVE.

This system is based on an ingenious spinner motor, which gives three speeds in either direction at maximum motor efficiency, control being had from two double-throw switches.

The motor consists of an ordinary *squirrel-cage* rotor on the main shaft. Concentric with the rotor and around it, adapted to revolve freely on extensions of the main-shaft bearings, is a spinner, provided with a brake. On its inside surface the spinner is wound to drive the rotor, while on its outside periphery the spinner has a squirrel-cage winding that enables it, in turn, to be driven as a motor by a surrounding stator built into the ship. Thus the machine consists of two motors concentrically arranged around a common axis, either of which may be made to revolve in either direction, or the spinner may be held by the brake while the rotor is revolving. The speed of the spinner, in this manner, may be added to or subtracted from that of the rotor, giving three speeds in either direction without the use of resistances in the rotor circuit, without varying the periodicity of the main circuit and without pole changing.

The relative speeds of the rotor and spinner are determined by the numbers of the poles in the spinner and stator.

In the case of a battleship similar to the *Florida* this method would contemplate a three-shaft arrangement, developing 30,000 electrical horsepower, at 150 r.p.m. The estimated loss in transmission in dynamos and motors is 10 per cent., which, it is claimed, would be more than compensated for by the gain in turbine and in propeller efficiencies.

The turbo-generators would have a speed of 1,200 r.p.m. The rotor revolves at 90 r.p.m., the stator at 60. The three speeds at which maximum efficiency would be obtained are, therefore, 30, 90 and 150 r.p.m., corresponding, respectively, to steerage way, to 10 knots and to 20.5 knots.

The equipment would weigh considerably more than an all-turbine arrangement. The overall diameter of motors would be 15 feet.

An installation of bituminous producer-gas engines, driving generators for this method of propulsion, it is reported, is to be installed on a cargo vessel on the Great Lakes during the present year. No reliable data concerning this installation have been received. Its development will be followed with great interest.

NUMBER OF PROPELLER SHAFTS.

Having definitely abandoned the single screw for all important vessels it is almost axiomatic that an arrangement of propelling machinery on two shafts is the best that remains, for reasons of simplicity, and of propeller efficiencies. But we are sometimes forced into a three- or four-shaft arrangement for reaction-turbine ships by the excessive length of engine room which otherwise would be required. This is not without advantages, however, in the ease of overhaul afforded by the smaller turbines and in the minimizing of the effect on the total propulsive power of the vessel of damage to one of the units.

As between three shafts and four shafts the latter are preferred for large vessels on account of the probability of better propeller efficiencies. Our lack of experience renders the design of the center propeller in a three-shaft arrangement rather difficult. The proximity of this propeller to the hull and the likelihood of its being robbed of its water by the wing propellers introduces an element of uncertainty as to its efficiency. It is probable that when properly designed and located three screws will give almost the same efficiency as two or four, yet there are startling instances of trials of triple-screw vessels in which the best speeds were obtained with the center screw idle. We have rejected the three-shaft arrangement for the new battleships, because with reaction turbines the four-shaft arrangement requires a shorter engine room. With impulse turbines the two-shaft design gives more room and greater simplicity on less weight, but in case we go to higher powers in our future battleships three screws will

probably be necessary in order to avoid excessively large units.

In the case of destroyers we use three shafts for Parsons turbines and two shafts for the Curtis and Zoelly turbines.

INTERNAL-COMBUSTION ENGINES.

Tests have recently been completed at the Norfolk Navy Yard of internal-combustion engines for launches. Nineteen representative makers submitted engines for the tests. Each engine was tested for four or five days in the shop, and an equal length of time in a 40-foot launch.

As a result of these tests, eight 4-cycle gasoline engines and one 2-cycle kerosene-oil engine were found suitable for naval use.

A gasoline engine to be sufficiently reliable for service use must be of heavy construction and securely mounted in a heavily-built boat. It has been conclusively demonstrated that a lightly-built high-speed engine, however carefully built and operated, will not do. A 4-cycle engine should weigh not less than 45 pounds per H.P., and the 2-cycle engine (for small powers) not less than 35 pounds per H.P. The revolutions should not be much greater than 500 per minute.

Designs are being prepared in the Bureau of Steam Engineering for standard gasoline engines for service use. It is probable that eventually the heavy-oil engine will displace the gasoline engine on account of the danger incident to keeping gasoline on board ship. The Department is bending every effort towards the development of an oil engine suitable for small boats.

The use of internal-combustion engines for propulsion of large vessels has long been an attractive subject to naval engineers. Such engines have shown marked economy in large units on shore with constant load when run by producer gas. We are often asked why we do not use such engines in naval vessels. The variations in load, the provision that must be made for reversing, and the necessity for a control of speed

sufficiently delicate for maneuvering in a marine installation, present difficulties that have not thus far been successfully met. There are difficulties, too, in connection with the design of a producer suitable for bituminous coal. The size of gas-engine cylinders suitable for marine work is at present limited to about 100 H.P. on account of temperature difficulties in larger cylinders. Thus, in the present state of the art, an installation for one of the new battleships would require 320 cylinders, and, needless to say, limitations of space alone would prohibit such an installation. With a gas engine, too, the cylinders are absolutely independent, one of the other; there is no mutual understanding, as with the cylinders of a steam engine, in which each cylinder is very much concerned with the performance of its neighbors. For this reason the problem of simultaneously reversing all the cylinders of a gas engine is a difficult one.

The British gunboat *Rattler* was in commission for about eight months with an anthracite producer, supplying gas to a 5-cylinder, 500-H.P. single-acting engine making 120 revolutions per minute, exhausting into the smoke stack. Speed could not be reduced below sixty-five turns. A hydraulic clutch was used for reversing; this wore badly and was unsatisfactory. The weight of the installation was greater than that of a steam plant of equal power.

At present, therefore, internal-combustion engines are unsuited for marine installations of greater than 1,000 H.P., and in small installations are justified only in special cases. There is, however, prospect of a considerable improvement in the design of bituminous producers, and a well-directed effort to overcome the difficulties which I have enumerated, so that the subject is still very much alive.

Our experience with internal-combustion engines in launches is that their maintenance is very expensive, amounting to 35 per cent. per annum of the original cost.

In the British and German Navies heavy-oil engines are used to drive a portion of the dynamo plant on some of the recent vessels. Such an arrangement permits the lighting

and ventilation of the ship when, for purposes of overhaul, all boilers are cold.

With us, however, this contingency is so remote that the use of internal-combustion dynamo engines has not been justified. Our principal navy yards are all capable of furnishing current to ships whose dynamo plants are disabled.

OIL-BURNING BATTLESHIPS.

Our experience with liquid fuel, while limited, has been sufficient to convince me that, considered from an engineering point of view, the expediency of its use to the exclusion of coal, in battleships, is clearly indicated.

The principal argument advanced against the use of liquid fuel, in large vessels, is the alleged difficulty and uncertainty of obtaining it in the seaports of the world. This is a strategical question, and belongs to you gentlemen rather than to me, but I will discuss it very briefly. New oil fields are being developed in many localities where until recently they were unknown. The Argentine Government has directed the installation of oil burning as an auxiliary in their battleships that are being built in this country, and in connection with this appears the statement that large oil fields have been recently discovered in Patagonia. California produces an excellent grade of fuel oil, but we are sending Pocahontas coal around the Horn for use in our vessels on the West Coast. I have seen the statement that the new oil fields in Oklahoma produced last year an amount of oil equal to twice the coal burned by the United States Navy during the same time.

The United States produces more than half the world's oil supply, and it is being shipped abroad for use as fuel in the vessels of foreign navies. It would appear that fear of a difficulty in obtaining oil should not seriously interfere with the adoption of this ideal fuel for use in United States naval vessels.

In time of war we must depend on the fuel stored at our

naval bases or carried in our cargo vessels; and it is easier to store and carry oil than coal. With the Panama Canal completed, oiling stations at our possessions in the Pacific, together with some oil-carrying vessels, would afford a certainty of fuel supply at least equal to that now provided for coal.

It should be remembered that all our future destroyers will probably be oil burners, and that their work is with the battle fleet. Thus we are already committed to the establishment of means of supplying oil for the destroyers at all bases at which it would be required by battleships.

That oil-burning battleships have not been more seriously advocated abroad is due entirely to the fact that, with the exception of Russia, no European nation has a supply of oil that can be depended on in time of war. It is logical, therefore, that our nation, producing, as it does, more than one-half the world's supply of fuel oil, should be the first to accept the great military advantages of oil-burning battleships. These advantages have been argued so often that I will not dwell upon them now. Briefly, with liquid fuel, we can:

(a) Reduce the weights and space required for boilers;

(b) Reduce the fireroom personnel;

(c) Eliminate ashes and cinders, and practically eliminate smoke;

(d) Increase the cruising radius.

The principal advantage, from a military point of view, of oil burning in a turbine-driven vessel, lies in the ability of such a vessel to be driven at full power with but slightly greater effort on the part of the personnel than at low speed, and in the fact that full speed can be maintained for periods limited only by the supply of fuel.

I wish, gentlemen, to place myself emphatically on record as advocating oil-burning battleships for our service.

IMPROVEMENTS IN ECONOMY BATTLESHIP FLEET.

In comparing the present coal consumption of the battleship fleet, which means its cruising radius, we will use as a standard that shown on the cruise of the battle fleet from San

Francisco to Hampton Roads, on which, as you will remember, a considerable improvement was shown over the first leg of the world cruise.

The average coal consumption per knot of the battle fleet at present, referred to a standard speed, is 6 per cent. better than that of the best ship on the world cruise.

The average coal consumption per knot of the battle fleet at present, at 12 knots, is practically the same as that at 10 knots on the world cruise.

Since the world cruise there has been an improvement in coal consumption per knot in the battle fleet of twenty per cent.

There is a similar improvement in the smaller vessels of the service.

The armored cruisers have apparently not improved, but are burning a little over 1 per cent. more coal per knot than last year.

In consumption of lubricating oil per knot there has been an improvement of 8 per cent. from December, 1909, to April, 1910, over that of the preceding year, for all vessels of the Navy. The average cost per knot of lubricating oil for all vessels is now 3.9 cents. This is not as good as the best merchant practice, but is constantly being improved. Our best ships cost less than 1 cent per knot.

As illustrating the ability of ships to maintain themselves, the cost of repairs due to casualties was four-tenths of a cent per horsepower during the last year. The cost of new machinery and boilers for naval vessels, included under the head of "navy yard repairs and changes," was 53 cents per horsepower during this period.

Including the power of the auxiliary machinery the average cost of developing one horsepower in the Navy is now about 2.04 pounds of coal per hour. This includes all expenditures of coal for whatever purpose, and also includes all power developed for whatever purpose. The figure is the average result as shown by the steam logs of vessels in service. The ratio between the average power developed and the maximum

power of naval vessels is extremely low as compared with any economical commercial practice. Considering this low-power factor the present result is not bad. The coal efficiency compares very favorably with that obtained under steaming conditions elsewhere. According to reports received, in one very large navy the cost in coal of developing a horsepower hour is 25 per cent. greater than in the United States Navy. These improvements in coal economy have been principally due to the following causes:

(a) The converting of evaporating plants from single to double effect. The power to distil fresh water sufficient for all purposes is a most essential feature in self-maintenance, and especially so if engaged in a campaign. The economy in making water simply means that the radius of action is increased accordingly.

The gain in economy from the use of the double effect is theoretically 48 per cent. Actual gains of from 35 to 40 per cent. are obtained. There is no appreciable reduction in the capacity of the plant.

(b) More careful and systematic firing, induced by analysis of smoke-pipe gases. Most of the larger ships have been equipped with apparatus for sampling smoke-pipe gases and determining the proportion of CO_2 therein. It is found that the percentage of CO_2 gives an accurate measure of the efficiency of combustion when coal is used as fuel. The principal furnace loss is the heat in the excess air which passes up the funnel. With a CO_2 percentage of 14 a condition of maximum efficiency is obtained in which there is no excess of air. Should this drop to 5 per cent., as was the condition in many of our ships when this apparatus was first installed, there is a loss in efficiency of 24 per cent., due to excess air.

In some of the ships a record of the CO_2 percentage of each fireman is kept, and the ability of the fireman is rated thereby. On one ship, where this system is followed, it was found that the number of buckets of coal required per watch varied progressively from 30.5, with a CO_2 percentage of 11.9, to 34.4, with a percentage of 9.0, the amount of steam generated remaining the same.

One of the first things learned by gas analysis was that our boiler casings leaked air. These have been made tight, with a corresponding saving in coal, reported in some cases to amount to as much as 8 per cent.

(c) The tightening of the pressure parts of boilers, particularly the bottom-blow valves. A new type of valve has been fitted on the larger ships during the past year, which removes a serious source of loss that has hitherto existed, due to the inability to keep the valves of the old type tight.

(d) More efficient propellers have been fitted on the *Kansas* and *North Carolina*, and are being made for the *Mississippi* and *Vermont*. As soon as accurate data on the efficiency of existing propellers under service conditions can be obtained, it is hoped to improve the efficiency of the fleet as a whole by replacing those propellers which are least efficient at cruising speeds.

(e) Greater attention to steam traps. It is found that leaky traps have in the past caused considerable losses.

(f) A more careful attention to auxiliary machinery, particularly to the condition of the valves and steam cylinders. In a battleship at cruising speed, one-third of all the steam generated by the boilers is used in the auxiliaries.

As the service in general fully realizes, smokeless combustion is greatly to be desired, and, while we have made some progress along this line, notably in the new coal-burning destroyers, still the problem has not been solved as yet. There are a number of smoke-consuming devices on the market, but we have been unable to find one that is even fairly efficient in a shore plant, and practically all are prohibitive in service installations because of confined space and high rate of combustion, and especially from the fact that nearly every one of them requires the expenditure of considerable steam for its operation.

FULL-POWER TRIALS OF BATTLESHIPS.

In Table IV are shown the results of the full-power trials of the battleships, held off Guantanamo last winter, as compared with their trial performances:

Name of ship.	Designed speed, knots.	Contractor's trial.			Fleet trial.		
		Speed.	I.H.P. main engines.	Displacement.	Speed.	I.H.P.	Displacement.
Connecticut	18	18.78	19,333	16,000	19.025	20,456	17,190
Kansas	18	18.09	19,302	16,000	18.960	20,273	16,890
Minnesota	18	18.85	19,896	16,000	18.420	20,685	17,200
Louisiana	18	18.82	20,443	16,000	19.073	19,793	16,692
New Hampshire	18	18.16	16,772	16,000	18.950	21,131	16,738
*Mississippi**	17	17.11	13,361	13,000	17.015	13,141	13,179
Idaho	17	17.14	13,765	13,000	17.291	14,017	13,400
Georgia	19	19.26	24,948	14,948	19.003	20,847	16,414
Nebraska	19	19.06	20,947	14,948	18.900	21,369	15,950
Virginia	19	19.01	22,501	14,979	19.054	20,805	16,140
Wisconsin†	16	17.17	12,322	11,565	16.810	10,828	12,142
Missouri	18	18.15	15,642	12,500	17.890	14,706	12,959
Vermont	18	18.33	17,824	16,000	15.140	8,900	16,968
New Jersey	19	19.18	22,805	14,948	18.460	19,251	15,280
Rhode Island	19	19.01	19,890	14,680	17.651	13,092	16,201

* One propeller injured by tug previous to trial.
† Safety valves set at 27½ pounds less than designed pressure.

The trials showed serious weakness in the feed pumps and blowers of the three last named ships. These weaknesses are being corrected.

With the exception of these three ships, the trials were most satisfactory. Omitting these vessels, the fleet, with an average displacement 830 tons greater than on contract trial, developed an average speed .449 knot greater than the designed speed, and .066 knot greater than the average contract trial speed.

On these trials the main engines withstood overloads amounting, in the case of one ship, to 40 per cent. in excess of the designed load.

An interesting feature, developed by the trials and shown in the table, is the improved propeller efficiency of the *Virginia* and *Louisiana*, whose engines were changed from in-turning to out-turning shortly before the trials. Unfortunately for the sake of comparison, the *Virginia's* present propellers differ from those with which she ran her contract trials. The *Louisiana's* propellers, however, are the same.

COMPARISON OF MEASURED-MILE COURSES AT ROCKLAND, PROVINCETOWN AND DELAWARE BREAKWATER.

The Board of Inspection and Survey for Ships has recently held exhaustive comparative trials of the *Michigan*, *Reid* and *Flusser* over the measured-mile courses at Rockland, Provincetown and Delaware Breakwater, to determine the effect of depth on speed.

The depth on the Rockland course varies from 330 to 480 feet; that at Provincetown from 150 to 160 feet, and at Delaware Breakwater from 135 to 150 feet.

These trials indicate that the *Michigan*, with 16,000 tons displacement and 24 feet 6 inches mean draught, requires about 1,900 less I.H.P. for a speed of 19 knots on the Rockland course than at the Delaware Breakwater, and somewhat less power for this speed at Provincetown than at the Breakwater. There is a corresponding effect in the case of destroyers. The *Reid*, displacing 700 tons, and with a mean draught of 8 feet 0½ inch, requires for a speed of 31 knots about 1,000 less S.H.P. at Rockland than at the Breakwater, and less at Provincetown than at the course in Delaware Bay.

REPAIR SHIP.

Our battleship fleet is, to a greater degree than ever before, self-sustaining as regards repairs to engineering material. This is evident in a decreased amount of navy-yard work. The machine-tool equipment of many of the ships has been increased to as great a degree as is possible within the limitations of space available. In some of the ships small cupolas capable of melting 100 pounds of brass and 60 pounds of iron have been improvised, and have proved very valuable for the manufacture of small castings not subject to much stress. A standard oil-burning furnace will be supplied to all the larger ships.

The repair ship *Panther* has rendered most valuable aid to the fleet, and has done much of the work heretofore done at the yards.

For times of war, however, when the ability of the fleet to be self-sustaining may be of the utmost strategic importance, our facilities for repairs are hopelessly inadequate. Realizing this, there have been prepared designs of a proposed repair ship of 13,500 tons displacement and 14 knots speed, with an equipment which will enable it to—

(a) Make patterns and castings up to four tons in weight. This includes all iron castings for small vessels, and, in case of a battleship, all except main cylinders and turbine casings. (These in any event would be repaired by a welding process.)

(b) Forge anything on board ship except main shafting;

(c) Roll and bend any plate that may be required for the hull or for boiler drums;

(d) Build up lengths of copper piping, and repair and tin defective piping;

(e) Do any required machining;

(f) Carry all materials for the prosecution of this work.

The proportion of foundry and forge equipment is such as to provide sufficient material for its own machine shop and for the machine shops of all vessels of the fleet. Thus, in emergencies, all the equipment of the individual ships, as well as that of the repair ship, could be run at full capacity.

The design of the repair ship provides for convenient and efficient handling of material progressively from its raw condition through the processes of casting, forging and machining in well lighted shops.

The acquisition of such a vessel is of vital military importance to the fleet in that it will, in time of war, enable the fleet to repair damages sustained in action to a sufficient extent to render it seaworthy and capable of renewing action. In time of war or peace it will render the fleet self-sustaining as regards all repair work to machinery, hull or ordnance equipments, except that necessitated in general refitting.

The experience of the world cruise of the battle fleet demonstrated rather conclusively that extensive cruising does not of itself contribute to deterioration of steam machinery,

provided the nature of the cruising is such that fixed periods for periodic overhaul are provided.

It has been too frequently the case that in arranging itineraries no attention has been paid to this point, and in many cases commanding officers hesitate at allowing disablement of the main engines for periods greater than twenty-four hours, although such disablement is required in the interests of an efficient maintenance of the plant.

An instance of this has recently developed in the case of two armored cruisers which require such extensive repairs in their engineer department as to render it necessary to lay them up for a period extending over several months. Until these repairs are completed these ships are without strategic value as a component part of the fleet. The present condition of the machinery of these vessels can best be described as generally run down. This is due in great degree to the nature of service to which they have been subjected—extensive cruising with improper spacing of overhaul intervals.

Undoubtedly, our most important duty as regards the machinery of the fleet is that of upkeep—the taking of the proverbial stitch in time which will prevent the accumulation of trouble in such quantities as to disable the fleet as a fighting unit.

This, however, requires coöperation such as is being accomplished under the present organization.

BOILERS.

The question of the type of boiler for future battleships is one that is receiving due attention. The present Babcock & Wilcox boiler is giving satisfaction, and that company has recently completed an exhaustive series of tests of a sample boiler for the *Wyoming* and *Arkansas*. In these tests they more than met every guarantee and, in fact, attained improved economies of evaporation at all rates of combustion.

The boiler stood a further test of forcing under 5 inches of air pressure, burning 70 pounds of coal per square foot of grate surface; all without signs of damage.

Notwithstanding all this, there is a growing sentiment favoring the small-tube express type of boiler for battleships, as giving more elasticity in forcing, especially in a turbine installation, as well as saving weight, on almost the same economy. The experiment station at Annapolis has been engaged in research work for more than a year, with considerable success, in an endeavor to determine proper methods and materials for the interior preservation of our boilers. This, of course, is quite essential if we adopt small or bent tubes in boilers.

In-swinging furnace doors have been fitted on all the battleships and are rapidly being installed in other naval vessels. The design of these doors is such as to cause them to automatically close in the event of a ruptured tube. Already, in the *Georgia*, they have prevented serious injury to the fireroom force.

Mosher boilers are being installed in the battleships *Kearsarge*, *Kentucky* and *Illinois*, to replace their Scotch boilers. The contract for these boilers calls for a guaranteed evaporation of 11¼ pounds of water per hour per pound of combustible at full power, when burning 40 pounds of coal per square foot of grate.

The difficulties heretofore existent with the reciprocating engines of forced-draft blowers have been overcome by the adoption of turbine-driven blowers for destroyers and motor-driven blowers for battleships.

Our new destroyers, propelled by turbines, with liquid fuel and turbine-driven fans, are provided with a machinery installation from which most of the frailties hitherto existent on vessels of this type are eliminated, and which should render them capable of running at full power at any time on demand, and for periods limited only by the fuel supply.

Turbine-driven centrifugal feed pumps are being developed and, although at present they are somewhat handicapped by excessive weight and space required, it is probable that these disadvantages will be overcome to a sufficient extent to allow us to show them in new designs. Experience with these

pumps in shore plants indicates that they are less prone to disablement than reciprocating pumps.

I think this covers the scope of my lecture.

In conclusion, I will say that engineering in our naval service is today in a position more nearly commensurate with its importance to the fleet than seemed possible a few years ago. This is due in part to the enthusiastic efforts of the younger line officers who have in recent years operated our machinery afloat, more to the legacy of experience, ability and good will from the old Engineer Corps, but is principally the result of an appreciation of the importance of engineering on the part of the older officers in the service, who, in responsible administrative positions, both in the Department and in the Fleet, under our able Secretary, are welding the personnel and materiel of engineering into the organization of the fleet in such manner as cannot fail to produce perfect homogeneity.

COMPARATIVE PERFORMANCE DATA OF RECENT DESTROYERS.

By Lieutenant Commander David F. Boyd, U. S. N., Member.

The destroyers of the 7th Division, Atlantic Torpedo Fleet, were commissioned and assembled at the Navy Yard, Charleston, S. C., on the following dates:

Name.	Where built.	Commissioned.	Arrived Charleston.
Flusser	Bath, Me.	October 28, '09	December 18, '09
Reid	Bath, Me.	December 3, '09	January 3, '10
Smith	Cramps	November 25, '09	December 10, '09
Preston	N. Y. S. B. Co.	December 24, '09	January 12, '10
Lamson	Cramps	February 10, '10	February 20, '10

After arrival at Charleston the destroyers began making twenty-four-hour coal-consumption runs, the resulting curves of which are shown in Plate I. Detached shaking-down cruises of about ten days' length were at intervals taken to Key West and return.

On March 21, 1910, the destroyers began cruising as a divisional unit, the distance steamed to date being about 9,500 miles, this being exclusive of numerous short runs. The standard speed has varied from 18 to 25 knots.

On March 24, 1910, the first competitive four-hour, full-speed run was made, on course from Key West to Pensacola, Fla. The Division stood out of the North-West Channel from Key West at 4:35 P. M., headed for Pensacola, and formed line (300 yards' interval) at 20 knots standard speed. At 6:00 P. M. the four-hour race began, the vessels then being approximately in latitude 25 degrees 02 minutes N., longitude

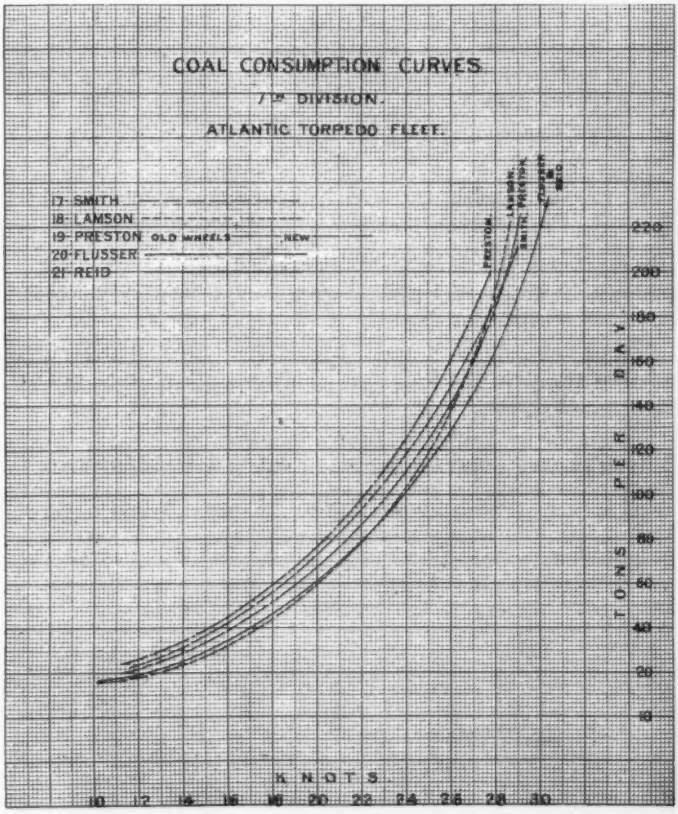


Plate I.

82 degrees 17 minutes W. The run was made in from 15 to 40 fathoms of water. Wind, N.W.; force, 2; sea, perfectly smooth. Displacement of destroyers averaged about 910 tons, 200 tons above trial displacement.

The trials ended at 10:00 P. M., in the following order, the vessels having made the following speeds, estimated from standardization curves, giving the speed corresponding to the average revolutions.

1. *Flusser*, ... 29.40
2. *Reid*, ... 29.80
3. *Smith*, ... 28.25
4. *Preston*, ... 26.95
5. *Lamson*, ... 24.52

NOTE.—Although the *Reid* apparently developed more speed than the *Flusser* (according to standardization curves), yet the *Flusser* actually gained a distance of 1½ miles, due, perhaps, to errors in standardization curves of both vessels, or to bad steering, or both.

Using data obtained on the builders' four-hour, full-speed trials as standard the following comparisons and remarks will be of interest.

COMPARATIVE DATA: BUILDERS' AND MARCH 24TH SPEED TRIALS.

Item.	*Flusser.*		*Reid.*		*Smith.*		*Preston.*		*Lamson.*	
	Builders'.	24th.	Builders'.	24th.	Builders'.	24th.	Builders'.	24th.	Builders'.	24th.
Draught of ship, mean	7-11½	9-6	8-0	9-5½	8-04	9-10½	8-02½	9-07½	8-02½	10-05
Corresponding displacement	686	890	690	882	716	910	719	930	690	934
Trim, inches	4½ by stern.	2 by stern.	6½ by stern.	3 by stern.	2 by stern.	6 by bow.	2½ by stern	Even keel.	2 by stern.	3 by stern.
Speed, average four hours	30.41	29 40	31.82	29 90	28.35	28.25	29.18	26 95	28.60	24.52
Revolutions, average four hours.	801	759	843	765	723	718	779	695	745	609
Main steam turbines, gauge	208	235	218	229	239	216	235	243	246	189
high pressure, gauge	186	210	201	211	199	196	175	179	189	150
Starboard L.P., gauge	25	30	32	33	45	35	27	25	41.5	23
Port L.P., gauge	25	30	33	35	45	35	25	25	42.5	23
Vacuum, gauge	28.4	27.6	28.6	28	25.8	27 2	27.4	27	27.2	27.3
Barometer	...	30.18	...	30.18	...	30.18	...	30.18	...	30.18
Main injection	56	68	58	68	70	68	56	68	54	68
Air-pump discharge	77	92	79	88	...	106	112	103	...	90
Overboard discharge	77	94	77	90	107	94	89	104	83	86
Temperature feed water	194	155	195	198	187	179	178	190	165	184
Back pressure auxiliary exhaust.	2.6	9.0	5.8	10.0	...	20.0	7.0	8.0	11.0	13.0
Air pressure firerooms	5.0	6.0	5.2	6.6	5.7	5 5	2.8	6.0	4.2	4.0

Condition of bottoms: Practically clean—all vessels.

The *Flusser*, *Reid* and *Preston* ran with cruising turbines disconnected; the *Smith* and *Lamson* with cruising turbines turning over in vacuo. The auxiliaries in use were the same, except on the *Smith*, where the dynamo was shut down, and on the *Lamson*, where No. 2 blower was shut down from 6:10 to 6:20, and No. 3 blower shut down from 6:20 to 10:00 P. M.

The vessels ran on approximately the following dummy clearances:

	M.H.P.	S.L.P.	P.L.P.
Flusser, ...	.013	.014	.010
Reid, no record, .	....	....	....
Smith, ...	.010	.016	.009
Preston, . . .	.015	.022	.019
Lamson, . . .	.009	.012	.019

The effect of displacement, on revolutions corresponding to chest pressures, and on speed corresponding to revolutions, is pronounced. Roughly, it is estimated in smooth weather, from cruising experience, about as follows, taking 20 knots (usual cruising speed) for comparison:

	Chest pressure, 700 T. I.P.C.	Chest pressure, 900 T. I.P.C.	Revs. for 20 kts., 700 T.	Revs. for 20 kts., 900 T.
Flusser, . .	120	135	465	482
Reid, . . .	126	141	472	490
Smith, . . .	130	140	470	485
Preston, . .	140	155	496	518
Lamson, . .	140	155	480	492

The following items are of interest on the five destroyers during this trial:

Flusser: At 7:30 P. M. the man on watch on the oil-return manifold gave the signal that something was wrong, and engines were slowed. Examination showed that he had just at that moment discovered that no oil was flowing from the I.P.C. forward bearing. This bearing had been, of course, cut out, the I.P.C. turbine being disconnected, and no oil had been coming from it at any time. The engines were put full speed ahead as soon as his mistake was discovered (at about 7:33 P. M.). A great quantity of water was lost by the overflowing of the feed tank, due to the fact that the column of water from the air-pump discharge could not pass fast enough through the filter compartments in the upper part of the tank. This was rectified subsequently by cutting a hole in

one filter compartment, thereby short-circuiting water to the bottom of tank.

Reid: At 8:20 a drain plug at the bottom head of water end of the main feed pumps, forward fireroom, blew out; and the tops of fires had to be killed in boilers "A" and "B" by flushing with fire hose. At 8:40, fires having been built up on these grates (using oil), the ship went ahead full speed. A steel plug had been driven in drain hole; but at 8:45 this plug again blew out, and fires had to be killed once more. The same steel plug was then driven in and shored up; the fires were again built up, and at 8:50 the ship was again going ahead full speed. Fires were carried from 14 to 18 inches thick and no difficulty was experienced in keeping full head of steam, the main high-pressure by-pass being open about one turn for one and one-half hours.

Smith: Nothing of interest on run.

Preston: The entire run of the *Preston* was seriously affected by main-feed troubles. It repeatedly became necessary to use cold feed, thus killing the steam. Examination later of the steam valves showed heavy scoring and heavy leakage from steam to exhaust faces, and that the babbitt rings of the water pistons were leaking.

Lamson: During the run two of the blowers ran hot most of the time and the oiling system carried away on one. The feed water could not be gotten above 190 degrees F. at any time. The vacuum on the starboard condenser was 27.07, on the port, 27.4, pumps running at full speed.

At Pensacola, Fla., the *Flusser* and the *Reid* ran their final acceptance trials on April 5, 1910, and April 6, 1910, respectively, making 29.51 knots and 30.21 knots, respectively, at 875 and 856 tons displacement, respectively.

The vessels of the 7th Division were docked at Charleston, S. C., the latter part of April and the first part of May. Here the wing propellers (diameter, 60 inches, pitch, 57 inches) of the *Preston* were removed and her spare propellers (diameter, 63 inches, pitch, 58 inches) were installed. This arrangement of the wing propellers (with a center propeller of, dia-

meter, 60 inches, and pitch, 57 inches) led at once to increased coal expenditure, as shown on the coal-consumption curve appended, marked "A." The Division then proceeded to Rocklaud, Me., and later to Provincetown, Mass., and Lewes, Del., where the *Reid* and *Flusser* ran standardization runs. The *Preston* standardized at Rockland, in view of the above-mentioned change in propellers.

The *Smith*, *Lamson* and *Preston*, on May 22, 1910, ran a four-hour full-speed run from Rockland to Provincetown, the *Smith* making her final acceptance trials on this run, averaging 28.80, 24.00 and 28.43 knots, respectively.

On May 28, 1910, the *Preston* ran her final acceptance trial in Massachusetts Bay, making 28.25 knots for four hours.

DIFFERENT TYPES OF DESTROYERS IN THIS DIVISION—TROUBLES EXPERIENCED—CHANGES MADE, ETC.

The engineering differences worthy of note bearing on economy and efficiency in these destroyers are shown in the table on following page.

Additional features which are not easily tabulated are as follows:

Flusser, *Reid* and *Preston:* High-pressure end of H.P.C. turbine faces aft. *Smith* and *Lamson* faces forward.

Have one double-beat maneuvering valve for each L.P. *Smith* and *Lamson* have one go-ahead and one astern valve for each L.P.

Flusser, *Reid* and *Preston* use exhaust steam on gland system, leak-offs to condensers. *Smith* and *Lamson* steam via reducing valves to glands, leak-offs to fifth expansion of L.P.

Flusser and *Reid:* Auxiliary exhaust to first expansion of port L.P., or third expansion of both L.P.'s or to third expansion of main; or to condensers. *Smith* and *Lamson:* Auxiliary exhausts to third expansion of L.P.'s or to condensers. *Preston:* Auxiliary steam exhausts to third expansion of L.P.'s or to condensers.

Flusser and *Reid:* Have no auxiliary condenser. *Smith*,

.....	*Flusser.*	*Reid.*	*Smith.*	*Lamson.*	*Preston.*
Type of boiler.....	Normand.	Normand.	Mosher.	Mosher.	Thornycroft.
Grate surface, ea...	86.8 square feet.	86.8 square feet.	92.1 square feet.	92.1 square feet.	87.5 square feet.
Heat'g surface, ea..	4,044 square feet.	4,044 square feet.	4,500 square feet.	4,500 square feet.	4,800 square feet.
Blowers.....	Bath, 7 in. × 5 in.	Bath, 7 in. × 5 in.	Forbes Sturtevant. 4½ in. × 5 in.	Forbes Sturtevant; 4½ in. × 5 in.	Forbes Sturtevant; 4½ in. × 5 in., girder foundation.
Main feed pumps...	Two Blake, simplex, 14½ in. × 10 in. × 12 in.	Two Blake, simplex, 14½ in. × 10 in. × 12 in.	Two Blake, simplex, 14½ in. × 10 in. × 12 in.	Two Blake, simplex, 14½ in. × 10 in. × 12 in.	Two Blake, simplex, 12 in. × 8½ in. × 16 in.
Location.....	One in ea. fireroom.	One in ea. fireroom.	Engine room.	Engine room.	Engine room.
Aux. feed pumps...	Two Blake, duplex, 7½ in. × 5 in. × 8 in.	Two Blake, duplex, 7½ in. × 5 in. × 8 in.	Two Blake, simplex, 14½ in. × 10 in. × 10 in.	Two Blake, simplex, 14½ in. × 10 in. × 10 in.	Two Blake, simplex, 12 in. × 8½ in. × 16 in.
Main feed pipe.....	4 inches.	4 inches.	6 inches.	6 inches.	4½ inches.
Air pumps.....	Two Blake, 10 in. × 22 in. × 15 in.	Two Blake, 10 in. × 22 in. × 15 in.	Two Blake, 10 in. × 22 in. × 15 in.	Two Blake, 10 in. × 22 in. × 15 in.	Two Blake, 12 in. × 26 in. × 18 in.
Condensers.....	Two Normand; expanded tubes; scoop 4,000 sq. ft.	Two Normand; expanded tubes; scoop 4,000 sq. ft.	Two, packed both ends, 4,013 sq.ft. each.	Two, packed both ends, 4,013 sq. ft. each.	Two, expanded one end, packed at other, 4,051 sq. ft. ea.
Feed-water heaters	Two, each 400 sq. ft. heating surface.	Two, each 400 sq. ft. heating surface.	One, 499 square feet heating surface.	One, 499 square feet heating surface.	One, 401 square feet heating surface.
Location.....	One in ea. fireroom.	One in ea. fireroom.	In engine room.	In engine room.	In engine room.
Circulat'g pumps...	Two, 3 in. × 3 in.	Two, 3 in. × 3 in.	Two, 7 in. × 6 in.	Two, 7 in. × 6 in.	Two, 8 in. × 8 in.
Oil coolers.....	One, 172 square feet cooling surface.	One, 172 square feet cooling surface.	One.	One.	Two, Koerting film type.
Propeller, diam.....	63 inches.	63 inches.	58 inches.	58 inches.	60 inches.
Pitch.....	58 inches.	58 inches.	60 inches.	60 inches.	57 inches.
Augmenter.....	Yes.	Yes.	No.	No.	No.

Lamson and *Preston:* Have auxiliary condenser. *Flusser*, *Reid*, *Smith* and *Lamson:* Have dynamo exhaust to auxiliary exhaust or to condenser. *Preston:* Has no dynamo exhaust to main condensers. *Flusser* and *Reid:* No relief valve or overflow to main feed pumps. *Smith*, *Preston* and *Lamson:* Overflow to reserve-feed tank.

The following small changes in machinery arrangements have resulted in increased efficiency:

Flusser.

(1) Installed both distillers in engine-room hatch, higher than the original position.

(2) Baffled evaporators.

(3) Substituted iron for copper washers under stud bolts connecting boiler casing to water drums.

(4) Installed water-service piping to main bearings.

(5) Installed suctions to trimming tanks.

(6) Installed additional baffle ring abaft main bearings to prevent leakage of oil, forced oiling system.

(7) Removed upper layer of brick in side walls of furnaces to prevent formation of soot pockets.

(8) Installed new valve (auxiliary exhaust) with square-sectioned socket to prevent the valve from rotating.

(9) Installed drain castings of larger internal diameter for oil drains to main bearings.

(10) Removed loofa from feed tanks and cut through feed-tank partitions.

(11) Port shaft worn $\frac{13}{1000}$ in all bearings except two, which were $\frac{4}{100}$; relined.

(12) Installed deck valve gear to blowers.

(13) Replaced thrust rings with others of a composition noted further on.

Reid.

(1) Baffled evaporators, and put in two circular discs to prevent priming.

(2) Installed another distiller and moved both distillers to a location in the engine-room hatch.

(3) Installed connection from trimming tanks to feed-water tanks and main air pump.

(4) Installed measuring tank from distiller to trimming tank, ship's tanks and reserve-feed tank.

(5) Installed brass baffle between oil-baffle rings on shaft and baffle abaft main bearings.

(6) Put in canvas strips around corners of bearing pockets to further prevent loss of oil.

(7) Installed water service from evaporator-feed pump to bearings.

(8) Removed packing from all pumps and substituted garlock packing.

(9) Put apron on all boilers from top of ash pan to furnace doors, to keep ashes and coal from blocking up holes in air casing.

(10) Reamed out rock shafts and put in larger taper pins in fire and bilge and auxiliary feed pumps.

(11) Had a new design of brick for furnace side walls manufactured, in order that top of lower drums between tubes could be readily cleaned.

(12) While steaming with four boilers on March 11, 1910, speed 25 knots, water was lost in boilers "C" and "D," and fires were hauled. The inner row of tubes in these boilers were found to be distorted, and a ruptured tube was found in each boiler. All tube joints in the steam drums were found to be leaking badly, and a few tubes were found to be leaking in the bottom drums. Several tubes were cut out, and the ruptured tube in boiler "D" was replaced; also, tubes were cut out to get at the ruptured tube. In boiler "C" the ruptured tube was plugged. To replace the damaged tube would have necessitated cutting out a large number of tubes. All tubes were rolled and the boiler subjected to a water test of 400 pounds. The boilers were found to be tight, and steam was raised on March 16, 1910. No leaks have developed since.

(13) Installed connections from trimming tanks.

(14) Changed exhaust leads from blowers to prevent water

in the exhaust from fire and bilge pump and auxiliary-feed pump from being forced back into the blower cylinders when the pumps are started.

(15) Installed oil manifold with larger discharge, to prevent loss of oil.

Smith:

(1) Installed new manhole plates, two-dog type, replacing the one-dog type.

(2) Added eight feet to smoke pipes to prevent gases from being drawn down into fire and engine rooms.

(3) Installed new distiller on deck.

(4) The original braces on the blowers were entirely too light. They were replaced with heavy angle iron, a ⅜-inch plate being bolted under the blower foundation. This greatly decreased the vibration. The forced-oil system was not efficient on account of heavy vibration, and an independent oil system was installed in each fireroom. This independent system is very satisfactory. The fan shafts above the fan were too light and had too little bearing surface. The shaft would bend just below the spider. To remedy this the shaft was cut off above the fan and a larger exterior shaft keyed on.

(5) The vibration of the blowers caused the blower-exhaust pipes to split and tear at the joint where the blower exhaust is brazed into the auxiliary exhaust line. An "S" has been put in the pipes and the vibration decreased so that no more trouble has occurred.

(6) A feed-water heater was fitted to the evaporators.

Lamson:

(1) Cut door and door-liner arches in two parts, so that in renewing these only the lower more frequently burned part has to be replaced.

(2) Blanked off discharge from feed-water heater drain to upper part of tank and connected the drain to the lower part of the feed tank, thus warming water in feed tanks 20 to 30 degrees higher than was attained before.

(3) Connected auxiliary condenser to the fresh-water tanks,

so the condenser can be used as a distiller, and the water run back to these tanks.

(4) Baffled both evaporators, thus increasing their capacity 20 to 30 gallons per hour. A separator is also to be put in the line between the evaporator and the distiller.

(5) Put a sloping wall in back of boilers, and have never had to repair them since, thus providing better support for back bricks.

(6) Connected evaporators to a small waste tank, so water can be tested before pumping it to reserve tanks.

(7) Put a snifting valve on salt-water side of auxiliary condenser.

(8) Connected main drain in after part of engine room to bilges.

(9) Trouble has been experienced with a Blake (Duplex) F. & B. pump; this trouble was finally traced to the tightness of the piston packing rings. This tightness caused the shearing of pins in the valve rocker arm.

Preston:

(1) Installed cast-iron liners and cheek plates in furnaces, thereby forming better support for front wall.

(2) Changed arrangement of bricking and form of anchor bolts, front wall, all boilers.

(3) Protected boiler casings (side) with mill-board secured and bounded by iron strips.

(4) Reamed out tight condenser-tube ferrules, to prevent them backing out when subjected to vibration.

(5) Installed both distillers on deck, well above original position.

(6) Baffled evaporators; changed size and lead of vapor-pipes; installed feed-water heater; installed small distiller discharge tank for aeration and testing of water.

(7) Installed suctions to trimming tanks.

(8) Installed distiller fresh-water-pump suction to reserve-feed tank and discharge to feed tank.

(9) Removed main feed-pump butterfly throttle and feed-tank float.

(10) Substituted soft packing rings for babbitt packing rings, main feed-pump water pistons.

SUMMARY.

In the *Flusser* and *Reid* the arrangement of scoops with small auxiliary circulating pumps were found to be inefficient for maneuvering. The installation of larger pumps has been authorized. A change in lead of exhaust piping in the fire-rooms has been made to prevent water being forced back into the blower-engine exhaust and thence into blower-engine cylinders, when pumps are started. This occurred on both these vessels, and was responsible for blower breakdowns. The composition of the thrust rings on the *Flusser* has been changed, trouble having been experienced with heating of the thrusts. The former rings were 6 parts copper, 1 part tin, ¼ part zinc, by weight. Without showing excessive heat —by thermometer or to touch—these rings ran warm and were worn as if ground down on an emery wheel, thus wiping off small particles which filled the oil grooves and were deposited in the forced-oiling system. These rings were replaced by others in which the ¼ part zinc was changed to ¼ part lead, and the trouble ceased.

On June 28, 1910, the *Flusser*, in a dense fog, struck the mooring chain of a large nun buoy with her port propeller. A crack extending along the upper thrust-bearing cap, L.P. turbine, was discovered soon after. There is a difference of opinion in the Division as to whether the shock of the propeller fouling the buoy could have caused this crack.

The *Smith* and *Lamson* have suffered from blower trouble, due to inadequate foundations, and to tearing away of brazing on blower exhausts, due to vibration; also to the angle at which the exhaust leads into the line. Considerable difficulty with leaky bulkhead stops has been experienced and finally overcome, as shown in the sketch appended (Plate II). On the *Smith*, while warming up and while the jacking gear was in, the thrust on the worm tore out its bed plate (the end of

the oil-pocket casting on the M.H.P. turbine casing). This was probably due to excessive vacuum while the jacking gear was in, or to large steam pressure accidentally used in warming up, or to both.

Eight feet has been added to the height of the *Smith's* smoke pipes, which relieves the great discomfort caused by smoke and cinders pouring down engine-room hatch. A similar change has been authorized on all the other vessels.

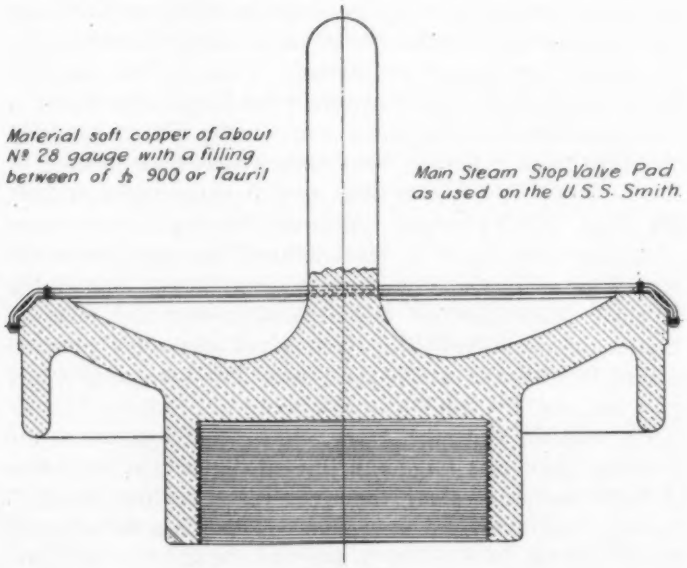

Material soft copper of about
№ 28 gauge with a filling
between of ♎ 900 or Tauril
Main Steam Stop Valve Pad
as used on the U.S.S. Smith.

Plate II.

The *Preston*, due to the excellence of the girder foundations, has escaped blower troubles. The main feed pumps have given trouble, due to the babbitt-metal rings on the water pistons. So long as the feed water is pure these rings seem to be all that is desirable; but so much trouble has been experienced from sand in the feed water, obtained from water boats, that these babbitt rings have been removed and

soft packing substituted. The *Preston* has on two occasions had condenser troubles, due to tight ferrules.

The wear of main bearings has been greater than the text-books relating to marine turbine installations lead us to expect. On the *Flusser*, for example, the wear experienced in six months' cruising has amounted to .013.

The practice of having steel yokes, valves and stems of such valves as are installed in and around the bilge, and on stops and checks, is considered to be a mistake; and these steel parts will have to be renewed with composition.

The installation of Class "A" steel bolts, studs and nuts in main steam lines has led to great annoyance and inconvenience whenever joints have to be broken. In some cases 50 per cent. of these studs have been sheared, although they had been graphited during the former making of the joint.

The use of loofa in the feed tanks has been discontinued. The loofa disintegrates and passes on through the feed system; and traces of it have been found in the boilers. The tube-sheets and tubes of the feed-water heaters have been found plastered with loofa. On the full-speed runs no chances were taken in having a system clear of this trouble, but the loofa was removed from the filter tanks.

The usual difficulty of leaky bulkhead stops has been remedied by a simple device designed by Chief Machinist Mate G. W. Keogh, of the *Smith*. Sketch of this device is appended (Plate II). This device is recommended for use in cases where leaky stops so frequently prevent repair work from being promptly undertaken.

DISCUSSION.

COMMANDER C. W. DYSON, U. S. N.—The coal-consumption curves furnished by Lieutenant Commander Boyd show a heavy decrease in economy for the *Preston* after the propellers had been changed.

As originally installed on this vessel all propellers were similar, according to the following particulars:

Each propeller:

Diameter, inches,	60.0
Pitch, inches,	57.0
Projected area, square feet,	12.27

On account of an injury to one of the outboard propellers both of these were replaced by the spares, the original middle propeller being retained. As thus fitted, the particulars of the propellers are as follows:

	Starboard and port shafts.	Middle shaft.
Diameter, inches,	63.0	60.0
Pitch, inches,	58.0	57.0
Projected area, square inches,	12.98	12.27

The starboard and middle screws are right-hand; the port screw is left-hand.

It is inherent with the three-shaft arrangement that the middle screw works at a disadvantage as compared with the outboard screws, on account of the much heavier wake.

A heavy wake, for high efficiency of propeller, entails increase in diameter, decrease in pitch and increase in surface over that required for propellers of equal efficiency working in a smaller wake.

While it is customary in destroyer design to make all three propellers alike as to pitch, diameter and surface, the arrangement does not tend to efficiency, and any further variation in the sizes of the propellers towards reducing the elements of the middle propeller below that of the wing screws will tend still more towards destroying the necessary synchronism which must exist between the three for maximum efficiency, and will still further reduce the impaired efficiency obtained when the three propellers are similar.

AIR PUMPS AND CONDENSERS AS USED IN NAVAL MACHINERY.

Lecture delivered before the School of Marine Engineering, U. S. Naval Academy, Annapolis, Maryland, April 29, 1910, by CAPTAIN F. H. BAILEY, U. S. Navy, General Inspector of Machinery for the Navy.

In the early days of steam engineering, when steam pressures were only a few pounds above the atmosphere, a good vacuum was of the utmost importance. As steam pressures increased, and our engines were speeded up, less benefit was derived from the vacuum. This is especially true of the high piston speeds used of late in marine engines, for it is impracticable to build the low-pressure cylinders large enough, or to provide sufficient port and valve area, to economically utilize steam at three or four pounds' absolute pressure. In fact, an examination of the indicator cards from almost any of our large ships at full power will show a back pressure in the L.P. cylinders from five to ten pounds absolute.

The introduction of the marine turbine allowed steam to be expanded down almost to the limit of pressure corresponding to the temperature of the condensing water, without greatly increasing its size, since it only required the addition of a few more rows of slightly longer blades and a freer exhaust to the condenser.

Condensing apparatus which was perfectly satisfactory for reciprocating engines that did not need a vacuum above 26 inches is not satisfactory for turbines, which can easily expand their steam down to 28 or 29 inches. That the gain due to a better vacuum is of great importance, even with high-pressure steam, can be easily shown theoretically from the laws of thermodynamics. The actual gain may not be

quite as great as the theoretical, but it is the increased economy due to the use of this better vacuum that makes the turbine so efficient. Without it but few turbines would be in use today for marine purposes.

The production of a good vacuum depends upon three factors.

First and most important: All air should be kept out of the condenser as far as it can possibly be done.

Second: The condenser should be so constructed that the air and steam, while sweeping over all the cooling surface, is brought to the air pump with the least possible loss of pressure, and at the same time is cooled to the temperature best suited for the vacuum desired.

Third: The design of air pump.

Air in Condenser.—The importance of keeping air out of the condenser by stopping every possible air leak, whether it be a joint, a pet cock or a stuffing box, can hardly be overrated. With the same vacuum and the same temperature the proportions of air and steam are constant; hence, the volume of this mixture to be handled by the air pump will vary directly as the amount of air present. Could all air be kept out the capacity of the air pump would need to be only sufficient to handle the condensed water. Do not forget that the capacity of the air pump, beyond that required to handle the condensed water, depends entirely upon the air leakage and not upon the horsepower of the engine, when the same vacuum is to be maintained at the same temperatures.

Air is harmful in another way. Experiment has demonstrated that the cooling surface is far less effective when much air is present. This appears to be due to the fact that the temperature of a mixture of steam and air is lower than that of saturated steam alone at the same pressure. This reduces the difference in temperature between the mixture of steam and air inside the condenser and the cooling water so that a less amount of heat is transmitted per unit of condenser surface. It is also probable that the cooling surface is always covered with a layer containing more air than the surrounding

steam due to air being carried along with the flow of steam against the cooling surface, where the steam is condensed and runs off as water, but the air not being condensed cannot escape so easily.

Arrangement of Cooling Surface.—With regard to the second factor, the exhaust should flow over the whole of the cooling surface with the smallest possible loss of pressure on its way to the air pump.

Air Pockets.—If there are any pockets or dead space in a condenser they become filled with air, and the cooling surface at that point is rendered practically useless. That part of the condenser becomes air-bound. This condition often occurs in feed-water heaters and evaporator coils, where a float trap permits the water alone to pass. On the trials of a certain scout cruiser, where the feed-water heater was fitted with a glass gauge, it was observed that the water was carried about the middle of the glass and above the drain. Under these conditions no air could escape, and the heater soon became filled with air instead of steam. The drain pipe, which was not lagged, was cold to the touch. That feed-water heater was ineffective because the man in charge of it did not crack the by-pass valve and let a little of the exhaust steam and air escape to the condenser.

Baffles.—The original condenser designed for the *Texas* received the exhaust at the middle of the top, and delivered it to the air-pump suction at the middle of the bottom with a suitable baffle between, making a satisfactory arrangement. In working out the arrangement in the ship the location of the air-pump suction was changed to one end without making any change in the baffling. On her first trials she could not get a good vacuum, for the steam followed the line of least resistance and took a short cut across one end to the air-pump suction. When the baffles were suitably arranged no further trouble was experienced.

Baffles cause increased resistance in the flow of exhaust steam through the condenser to the air-pump suction. They are usually necessary, but should be used as sparingly as

possible. We must always bear in mind that it is the vacuum at the turbine or engine which counts and not that at the air pump. The exhaust pipe from turbine to condenser should be large, the nozzle on condenser well spread out over the tubes, and the tubes themselves, where the exhaust steam enters, should be widely spaced, with open lanes extending into the tube nests so that the great volume of exhaust steam can come in contact with as much tube surface as possible when it first enters the condenser. A large part of the steam will then be condensed by the first rows of tubes, reducing the volume so much that the remainder can easily flow between the more closely-spaced tubes. All the steam should, so far as possible, pass over the same amount of tube surface, so that the condensed water and air will have a uniform temperature when it is collected at the air-pump suction.

At this point we will find the coldest steam in the condenser and likewise the greatest percentage of air in a given volume, hence the air pump will remove the greatest quantity of air per stroke.

TABLE I.—DATA FOR CURVES ON PLATE I. VACUUM = 26.28 INCHES.

Temp. in condenser.	Pressures.			Volume, cubic feet.			Per cent., by volume.	
	Steam.	Air.	Total.	Steam.	Air.	Total.	Steam.	Air.
122.8	1.823	.000226	1.823	676,800	84	676,884	99.988	0.01241
122.0	1.785	.038	1.823	3,946	84	4,030	97.92	2.08
120.0	1.689	.134	1.823	1,059	84	1,143	92.65	7.35
117.0	1.552	.271	1.823	481	84	565	85.13	14.87
114.0	1.426	.397	1.823	301	84	385	78.18	21.82
111.0	1.308	.515	1.823	213	84	297	71.72	28.28
110.0	1.267	.556	1.823	191	84	275	69.50	30.50
108.0	1.200	.623	1.823	162	84	246	65.85	34.15
105.0	1.098	.725	1.823	127	84	211	60.19	39.81
100.0	.946	.877	1.823	91	84	175	52.00	48.00
90.0	.696	1.127	1.823	52	84	136	38.24	61.76
80.0	.506	1.317	1.823	32	84	116	27.58	72.42

Design of Air Pump.—Before taking up the air pump itself, I will direct your attention to curves (Plate I) showing

the reduction in the volume of steam and the increase in the percentage of air, as it is cooled while passing through the condenser. The performance of the starboard engine of the *North Dakota* during her four-hours full-power trial has been selected for this purpose. The curves are plotted from the data given in Table I.

The engine was working up to 14,345 I.H.P., with a vacuum of 26.28 inches, or 1.823 pounds absolute pressure, and a hotwell temperature of 104 degrees F. The capacity of the air pump for each double stroke was 20 cubic feet, or at the average speed of 14.87 double strokes, about 298 cubic feet, per minute. Allowing a loss of pressure in the pump of one-fourth inch for lifting the foot valves and flowing through the layer of water over them, gives an absolute pressure in the pump of 1.7 pounds. Assuming that the temperature inside the air pump is the same as that of the hotwell, an assumption justified by the large amount of water present, we find that the pressure due to the steam is 1.066 pounds, and to the air, .634 pounds. Allowing 15 pounds of steam per I.H.P. per hour, the volume of water passing through the air pump is 57.38 cubic feet per minute, or in percentage we are pumping—

Steam, ...	50.64
Air	30.11
Water, ...	19.25

Making the proper corrections for the higher pressure in the condenser, the volumes pumped out of the condenser per minute become, in cubic feet,

Steam, ...	141
Air, ...	84
Water, ...	57

The volume of steam entering the condenser per minute at an absolute pressure of 1.823 is, in this case, 676,800, and of the air, as we have already found, is 84, making a total of 676,884.

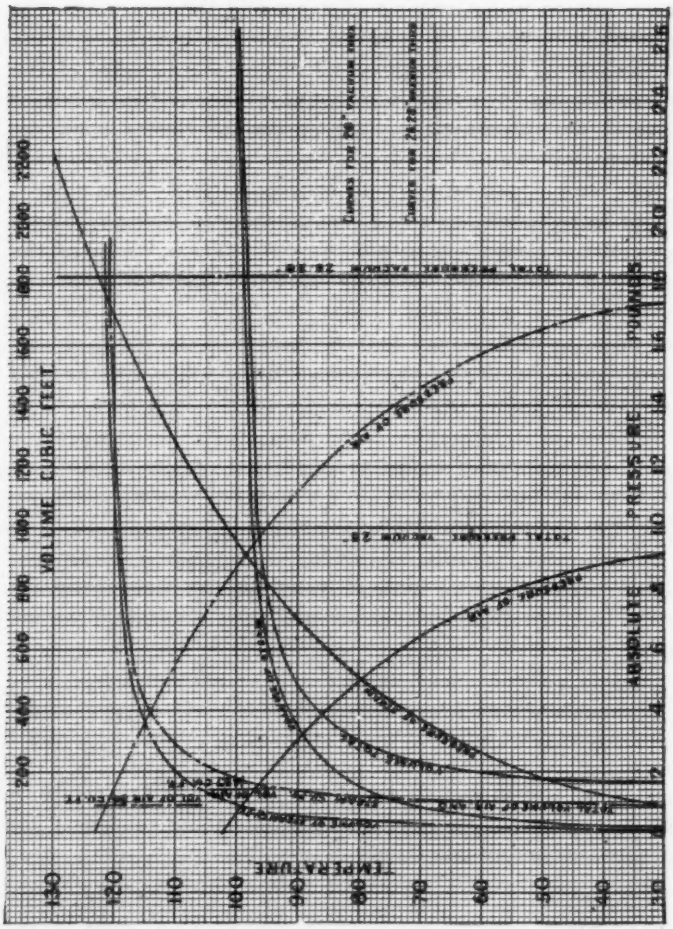


Plate I.

The proportion of air at first is so small that the mixture, if I may be permitted to use the term, acts like pure steam, and you will see, from an examination of the chart, that lowering the temperature only .8 of a degree reduced the total volume from about 677,000 to only a little over 4,000

cubic feet, and increased the percentage of air present from .012 to over 2. Lowering the temperature 12 degrees further, or to 110 degrees, decreases the volume to 275 cubic feet, and the percentage of air is increased to over 30. Here you will observe that the direction of the curve is changing very rapidly, and the total volume which in each case represents the total quantity to be pumped out, or the required capacity of the air pump for that particular temperature pressure and volume of air, now begins to decrease much more slowly.

Referring again to Plate I, the lower set of curves are drawn in the same way for the same actual amounts of steam and air. The vacuum, however, is only 28 inches, or one pound absolute, and the curves are constructed from the data given in Table II.

TABLE II.—DATA FOR CURVES ON PLATE I. VACUUM = 28 INCHES.

Temperature in condenser.	Absolute pressures.			Volume, in cubic feet.			Per cent., by volume.	
	Steam.	Air.	Total.	Steam.	Air.	Total.	Steam.	Air.
101.84	1.000	.0001228	1.0	1,195,417	145.7	1,195,563	99.988	0.01228
100.0	0.946	.054	1.0	2,552	145.7	2,698	94.6	5.4
97.0	0.864	.136	1.0	926	145.7	1,072	86.4	13.6
94.0	0.788	.212	1.0	541	145.7	687	78.8	21.2
91.0	0.718	.282	1.0	371	145.7	517	71.8	28.2
88.0	0.654	.346	1.0	275	145.7	421	65.4	34.6
85.0	0 594	.406	1.0	213	145.7	359	59.4	40.6
82.0	0 540	.460	1.0	171	145.7	317	54.0	46.0
79.0	0.449	.511	1.0	139	145.7	285	48.9	51.1
76.0	0.443	.557	1.0	116	145.7	262	44.3	55.7
70.0	0.363	.637	1.0	83	145.7	229	...	...
60.0	0.256	.744	1.0	50	145 7	196	...	...
50.0	0.178	.822	1.0	31	145.7	177	...	...
40.0	0.122	.878	1.0	20	145.7	166	...	...
32.0	0.089	.911	1.0	14	145.7	160	...	...

The two curves of total volume show the enormous increase in the size of the air pump required to raise the vacuum from 1.823 pounds absolute to 1 pound, or from 26.28 to 28 inches when the same amounts of air and steam are present at the same temperatures. Since the temperature of the injection water is raised from 20 to 30 degrees at full

power the lower curves show how difficult it would be to get a vacuum of 28 inches unless the air entering the condenser was reduced to a minimum.

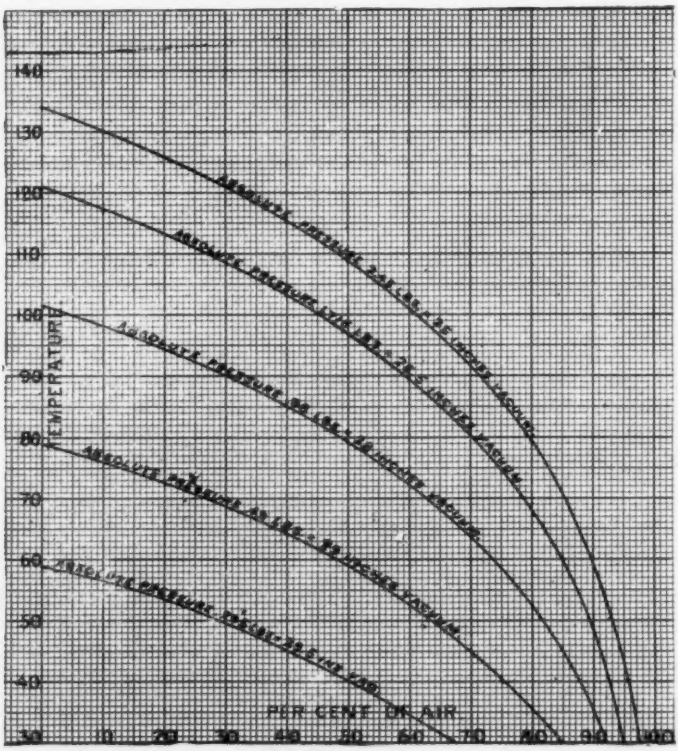


Plate II.

Percentages of Air and Steam.—Plate II shows a series of curves giving the percentages of air at given temperatures for different condenser pressures. These are constructed from data corresponding to that in the last two columns of Tables I and II. With the aid of curves of this form it is easy to determine the percentage of air and steam entering the air

pump at any time when we know the temperature and pressure. I most strongly advise you to always fit a thermometer and good mercurial vacuum gauge to the air-pump suction pipe, but the greatest care must be taken to locate these instruments so that you can obtain correct readings. The vacuum gauge must be located so that the pipe leading from it will always be drained, for a pocket in which water can collect may give very unreliable readings.

Another feature to which I would like to call your attention is this:

You will notice from the curves that almost all the steam is condensed with the first small reduction in temperature. Now, if the condenser is arranged so as to permit it, and a wet-vacuum pump is used, this water can be drawn off and some heat saved. The remainder can be cooled to the lowest practical limit and the work of the air pump reduced accordingly, without materially lowering the mean temperature of the total condensed water.

Effect of Clearance.—Clearance in air pumps is detrimental unless it can be entirely filled with water, which is seldom done. The water and steam are too intimately mixed to separate in the short time allowed at the end of the stroke. The effect is to reduce the efficiency of the pump, or the amount of vacuum which it can produce; thus, with no clearance the limit would be a perfect vacuum; with the clearance filled with water it would be that due to the temperature of the water. With a clearance equal to C, a volume at the beginning of the stroke equal to V, the pressure of the steam due to its temperature p, and the total pressure at discharge P, the limit of the vacuum produced will be

$$\frac{(P-p)\,C}{V} + p.$$

For example, let the clearance $C=\frac{V}{20}$, the pressure of the steam be two inches of mercury and the pressure of the discharge 17 pounds absolute, or 34 inches of mercury, the

limit of the vacuum which we could produce would be $\frac{(34-2)V}{20V} + 2 = 3.6$, or 26.4 per gauge. Since clearance is so harmful, several methods have been devised to reduce or counteract its effects as much as possible, or eliminate it altogether. The most common way is by virtually compounding the pump as usually done in the ordinary single-acting pump fitted with both bucket and foot valves. In the single-acting pump the above calculation represents only that part of the pump between the bucket and delivery valves. The action of the part between the foot and bucket valves would be the same, only the pressure of discharge would be very much lower, rendering the effect of clearance very much less.

Allowing the same ratio of clearance as above, we would have for that part of the pump between the foot and bucket valves $\frac{(3.6-2)V}{20V} + 2 = 2.08$ absolute, or 27.92 by gauge. It will be noted that in this case the limit is within .08 inch of what could be obtained without clearance. This shows why so much better results are obtained from vertical single-acting pumps than from the usual horizontal double-acting pumps used with our dynamo and auxiliary condensers.

Double-acting Air Pumps.—For increasing the efficiency of double-acting air pumps a method suggested by Professor Hollis, of Harvard University, while he was yet an Assistant Engineer in the Navy, was first used on the U. S. S. *Hartford.* It consists in connecting the clearance spaces at each end of the barrel of the pump by meaus of ports cut through the liner. The piston overruns these ports at each end by about half an inch. The effect is to put the two ends of the pump in communication and allow the pressures to equalize before the piston returns. The pressure behind the piston is reduced to almost that of the condenser, so the foot valves begin to open almost as soon as the piston begins to return. It has the same effect as reducing the clearance. The same principle is worked out in a different way by the Alberger Condenser Co. In this case it is accomplished by the use of

an equalizing port in the body of a mechanically operated valve which, when the piston is at the end of the stroke, connects the two ends of the cylinder, thus allowing the pressure to equalize.

Another feature which demands attention is the pressure required to lift the foot valves and force the air and steam through the water in the bottom of a vertical pump, especially if a high vacuum is to be carried. The steam pressure in the air-pump cylinder, while it is being filled, will be that due to the temperature of the water in the clearance space, or practically the same as that in the condenser, but the total pressure will be less than that in the condenser by the amount lost in lifting and forcing it through the foot valves and water over them.

Illustrating by example: Assume that the vacuum to be carried is 28 inches, or 1 pound absolute, and the temperature of the water is such as to produce a pressure of $\frac{6}{10}$ of a pound, about 85 degrees, leaving the pressure of the air $\frac{4}{10}$. Now, if $\frac{1}{10}$ of one pound pressure is lost in passing through foot valves, the total pressure in the pump will be $\frac{3}{10}$ of a pound, of which $\frac{6}{10}$ is steam and only $\frac{3}{10}$ air. The amount of air discharged will be only 75 per cent. of what it would have been had the air and steam entered freely; or, in other words, we would have to increase the volume of our air pump by 33 per cent. to maintain the same vacuum. One-tenth of a pound corresponds to a little over $2\frac{1}{2}$ inches of water, an amount no greater than is usually present in a vertical single-acting air pump.

I know of but two methods of avoiding foot-valve losses. One is to use mechanically-moved valves, and the other is to omit them entirely. Mechanically-moved valves are used almost exclusively in dry-vacuum pumps. The objection to them includes the increased complication, weight and space, that must be greater, since it becomes necessary to add a crank and fly wheel to the pump.

Foot valves may be omitted if we allow the piston to run over slots in the pump cylinder at one end, making a single-

acting pump, like the Edwards, for example; or in the center, making a double-acting pump, of which the feather-weight and Bailey air pumps are examples. This last method has the advantage of reducing clearances, as well as eliminating foot valves.

Most Suitable Type of Air Pump.—We now come to the question of what style or combination of air pump is the most suitable. Like almost any other machine, no one style of pump is best for all purposes. When pumps are attached to main engines built for moderate speeds, the vertical single-acting pump cannot be much improved. The Edwards is also used to a considerable extent, and gives probably about the same vacuum as the single-acting pump, but having fewer parts, it is less liable to breakdowns and easier to overhaul. It is also capable of running at higher speeds. For high-speed engines the Bailey pump, such as we use on our torpedo boats and launches, appears to fulfill the requirement better than any other kind. Being double-acting, it has about twice the capacity of the single-acting pump of the same diameter and stroke, and consequently is much smaller and lighter. Like the Edwards, it has no foot valves, but, unlike the Edwards, the delivery valve covers the whole end. This enables clearance to be reduced to a minimum and the pump to be run as fast as the high-speed engine to which it is attached.

Pumps without foot valves require more power to operate than those with foot valves. The reason is this: the pressure on the discharge side of the pump piston is the same for all pumps, but on the suction side it is different. In the ordinary pump this suction side is connected with the condenser, and the poorer the vacuum the greater is the pressure; but in pumps without foot valves the pressure is very much less than that in the condenser until near the end of the stroke, when this space becomes connected with the condenser. With a poor vacuum this might amount to a pound per square inch, or more, but with a high vacuum it is decreased so as to be of little importance. All, except dry-air

pumps, are liable at times to get almost solid water from the condenser. With attached pumps and, to a less extent, with crank and fly-wheel pumps, this brings a great strain on the whole pump and is one of the chief causes of breakdowns. The best remedy is to arrange the condenser and piping so that the water will flow steadily to the pump, thus preventing any accumulation of water when the pump is in operation. In the Edwards pump the discharge valves can be made so large that there is little danger of it being overloaded. In the Bailey pump, as the valve covers the whole end, there is practically no danger to the pump, when well designed. It is doubtful, however, if in either of these pumps at high speed the cylinder can fill with water in the short time the slots are uncovered. In the usual single-acting pump the area through the bucket valves should be made as large as possible. The velocity of flow ought not to exceed 1,000 feet per minute. If the bucket is not large enough to admit of so low a velocity, then the stroke should be shortened and the diameter increased. Foot valves should have the same area as bucket valves, but delivery valves may have a larger area with advantage. A large air chamber should also be provided over the delivery valves of all attached or crank and fly-wheel pumps, and a large pipe led to the feed tank.

When independent pumps are used with reciprocating engines it is largely a matter of choice between ordinary twin air pumps and the feather-weight type. Feather-weight pumps placed side by side but separately driven, as on the *Birmingham*, take up no more space than the ordinary twin-air pump. Since there are two virtually independent pumps in this case, either of which will produce a fair vacuum, there is much less danger of having to stop the engine on account of air-pump breakdowns.

At one time a number of crank and fly-wheel pumps were in use in our Navy. Some of these worked well, but most of them were so poorly designed that they were replaced by the direct-acting twin type. They were difficult to start, whenever they stopped for any reason. They were heavy and

occupied much space. About the only thing to be said in their favor was that they were fairly economical in the use of steam. With reciprocating engines some heat might be saved by using a wet-vacuum pump to take care of the water and a dry pump to take care of the air, but it is very doubtful if the gain would compensate us for the extra complication.

With a turbine installation a wet and dry-vacuum pump or some other equivalent device is a necessity, if we expect to obtain a high vacuum economically, for, owing to the lower absolute pressure in the condenser, the volume of air and steam which has to be removed is greatly increased.

Augmenter System.—As the Hon. C. A. Parsons has built more marine turbines than all other builders combined, it is reasonable to assume that he has given the subject much study. We will take up his system first, especially as it is worked out on lines which I believe to be correct. As you know, the augmenter takes the place of the dry-vacuum pump. The augmenter consists of a small condenser into which air and steam, drawn from the main condenser near the air-pump suction pipe, is forced by means of a steam jet. In this small condenser the steam and air are compressed and cooled at the same time. While the increase in total pressure may not be more than 1 or 2 inches of mercury, the pressure of air alone must have been increased that amount plus whatever is due to the temperature of the steam having been decreased. We have thus increased the percentage of air several times. This air is now discharged into the air-pump suction near the air pipe itself, and is prevented from flowing back into the condenser by means of a water seal. The air and condensed steam are then removed by the ordinary air pump. The important point here is that, no matter how much you increase the total pressure, so long as the temperature is not allowed to rise, the increased pressure is due entirely to the increased air which has been concentrated in this way.

Weir Uniflux System.—Weir, in his recent design of air pump, which we will use in some of our new battleships, accomplishes this result in another way. His air pump consists

of two air cylinders worked by a beam in the usual way. Each cylinder is a complete single-acting air pump of the usual type. One cylinder is used as a wet and the other as a dry-vacuum pump. The dry pump takes its steam and air from the top of the air-pump suction pipe and discharges it into the wet-vacuum cylinder above the bucket valves through valves opening downwards in the delivery-valve plate. As this discharge takes place during the down stroke of the wet pump the pressure is low. The air and vapor are cooled, I believe, by means of a spray of cold, fresh water admitted into the pump. Since the pressures are never high the dry-vacuum pump has but little work to do and it throws but little additional work on the wet-vacuum pump.

The principle in both the above systems is to compress and cool a large volume of steam and air into a much smaller volume, containing all the air but only a portion of the steam, thus allowing it to be handled with a much smaller pump.

Separate Wet and Dry-Air Pumps.—The U. S. Scout *Salem* has an arrangement similar to that successfully used in many shore installations. It consists of a centrifugal wet-vacuum and a reciprocating dry-air pump. The wet-vacuum pump is a turbine-driven two-stage centrifugal pump. This pump removes all the water of condensation from the condenser and delivers it to the feed tank. As the water flows by gravity into its suction it primes itself and cannot get air bound. It delivers the water steadily as it flows into the pump so long as it is kept running at a fixed speed. The dry-air pump is driven by a single-cylinder steam engine with fly wheel. The steam cylinder is 10 inches and the air cylinder 24 inches in diameter, with a common stroke of 18 inches. The air cylinder is water-jacketed for cooling purposes. It has mechanically-operated flat slide valves. Since this receives the air and vapor at a pressure of about 1 pound absolute under ordinary conditions and delivers it at 15 pounds, the load on the piston is 15 times as much at the end of the stroke as at the beginning. Like all crank and

fly-wheel pumps it is liable to stop if run very slowly, and when stopped cannot always be quickly started.

You now have a general description of three methods of removing steam, air and fresh water from turbine condensers. Let us now make a comparison of these different methods and see which seems to be best fitted for our purpose.

In making this comparison there are four things to be considered:

1st. Reliability.

2d. Efficiency of the apparatus in producing a vacuum.

3d. Economy in steam consumption.

4th. Weight and space occupied.

Reliability or freedom from break-downs or derangement, or the certainty that the apparatus will not stop at a critical time, is of the first importance.

In the first or Parsons system the augmenter or dry-vacuum pump is about as simple and has as few chances of going wrong as anything that can be found on board a ship. It may, in fact, be regarded as thoroughly reliable.

The wet-vacuum pump is exactly the same as an ordinary air pump, and, therefore, just as reliable. Should the augmenter fail the wet-vacuum pump would still continue to produce just as good a vacuum as usual with an air pump working in the ordinary way. The vacuum would probably not fall off more than an inch or two at the outside.

In the second or Weir's system both the wet and dry-air pumps are similar in design to the ordinary air pump, and should be no more liable to breakdown. The failure of the dry-air pump would still leave the wet-air pump in operation, so that a fair vacuum would still be maintained, but the failure of the wet-air pump alone would render the whole apparatus useless.

In the third case, which is represented by the *Salem's* installation, where the wet and dry-air pumps are separate units, the failure of either renders the other useless, for the dry-air pump cannot pump water and the wet-air pump cannot produce much of a vacuum, although I was informed by

the engineer of the Kent Avenue power station in Brooklyn, N. Y., that their centrifugal wet-air pump alone would maintain a vacuum of 16 inches. The centrifugal pump itself is not very likely to get out of order, and, when turbine driven, should be quite reliable. The ordinary reciprocating pump could be used if preferred. In either case, the pump must be placed below the condenser and the suction pipe led so that the water will flow by gravity into the pump, so that there will be no danger of it becoming air-bound. The dry-vacuum pump, with its mechanically-moved valve and its crank and fly wheel, if made large and strong, would not be likely to give trouble. You may ask why a crank and fly wheel are necessary. Dry-vacuum pumps, which are always double-acting, must be made with as little clearance as possible, since there is no water present to fill the spaces, and a fixed length of stroke is necessary to accomplish this result. Mechanically-moved valves are also desirable, and can best be driven and controlled from the crank shaft. The fly wheel is necessary to give a more uniform motion, and a crank and fly-wheel pump is more economical in the use of steam.

With a single air cylinder, as used on the *Salem*, the pump receives the air and steam at a pressure of, say, 1 pound, and discharges it into the atmosphere at a pressure of about 15 pounds absolute. This is over a very wide range, and not only tends to make the pump work irregularly, but it also heats the air and steam due to the compression, thus increasing their volume and also the work required to run the pump.

I believe that better results would be obtained by compounding the pump, that is, using two air and two steam cylinders, with the cranks at right angles. The first pump cylinders would compress the air and steam to four or five times its initial pressure, and the second cylinder would finish the compression. In a receiver between the two cylinders a cooling coil could be placed which would condense most of the steam, thus reducing the volume and the total work considerably. It should make a smooth-running pump and one not liable to stop. Since the compression would be but $\frac{1}{6}$ or

$\frac{1}{7}$ as much in the first cylinder the loss from clearance would be very much less than when only a single cylinder is used. With the air leakage constant, so long as the condenser supplies the air pump with air and steam at a certain temperature, the volume of air and steam so supplied will depend entirely upon the pressure. The lower the pressure or the better the vacuum the larger will be this volume, for, the temperature being constant, the pressure due to the steam will be the same, and the pressure due to the air will be the difference between the pressure of the air and the absolute pressure or vacuum produced by the pump. Now, the volume of any given quantity of air will vary inversely as its pressure; therefore the efficiency of any pump, so far as the production of a vacuum is concerned, depends entirely upon the volume of air and steam which it removes from the condenser. To illustrate by an example, suppose the temperature is 102 degrees, the pressure due to the steam alone would be 1 pound absolute. Now, in maintaining a vacuum of 26 inches, or 2 pounds absolute, half of the volume pumped would be air and half steam; or, to pump out 1 cubic foot of air at 2 pounds' pressure we would have to also pump out 1 cubic foot of steam, making 2 cubic feet in all. To maintain a vacuum of 27 inches, or $1\frac{1}{2}$ pounds absolute, with the same temperature, the steam pressure would be 1 pound, as before, but the pressure of the air would be but $\frac{1}{2}$ pound, or $\frac{2}{3}$ of the volume would be steam and $\frac{1}{3}$ air. Since the air is but half as dense as before, we must pump out 2 cubic feet to get the same weight of air as before, and with it we must pump out 4 cubic feet of steam, or 6 cubic feet in all. In other words, we require three times the air-pump capacity in this case to gain 1 inch in our vacuum.

It is now quite evident that the apparatus which handles the greatest volume of steam and air as it comes from the condenser will give the best vacuum, and, in my opinion, the choice in this respect lies between the augmenter and the crank and fly-wheel dry-vacuum pump. Unfortunately, as regards the augmenter I know of no experiments which show

how much air and vapor an augmenter of a given size does handle, and so cannot make any real comparison. We know that a steam jet is not an economical way of producing pressure, but in this case the pressure does not have to be increased very much to concentrate the air, since steam is being condensed in the augmenter at the same time. Until some one attaches thermometers and mercurial vacuum gauges to the main condenser and wet-air pump connections of the augmenter condenser, we will never know just how much the density of the air is increased in passing through the augmenter.

Such experiments are well worth trying, and I hope sometime some of you will not only have the opportunity but will take advantage of it and publish the results in the Society's journal. I think there is no question but that equally as good, if not better results, can be had with a well designed dry-vacuum pump, but unless the results are better the augmenter is the simpler and, therefore, preferable. The Weir system is well worked out and it should give better results than the ordinary air pump alone, but owing to its slow speed the volume of air and steam which it can handle is less than with either of the other systems. Independent air pumps of the ordinary type are very wasteful of steam. This is due to the fact that the highest pressure in the pump is at the end, thus requiring a cylinder full of steam at maximum pressure for each stroke. It seems to be a necessary evil for this type of pump, but the pump as a whole gives such good satisfaction that few of us would care to change, since crank and fly-wheel air pumps have proven so unsatisfactory in general. As the trouble with crank and fly-wheel pumps was due chiefly to the irregular flow of water through the pump, this objection does not apply to a dry-vacuum pump, and a great increase in steam economy can be obtained by using a crank and fly wheel. In the scout-cruiser trials the main air pump of the *Chester* used more steam than both the wet and dry-vacuum pumps of the *Salem*, and I feel quite sure that the separate wet and dry-vacuum pumps will always be found the

most economical combination so far as steam consumption alone is concerned.

With regard to space and weight much depends upon the care with which the different designs are worked out, but it is probable that the separate wet and dry-vacuum pumps would be heavier and require more space.

SCREW PROPELLERS.

(Continued.)

By Commander C. W. Dyson, U. S. N., Member.

THE INFLUENCE OF TIP SPEED OF PROPELLER BLADES ON PROPELLER DESIGN.

In the paper prepared by the writer and published in the August, 1910, number of this journal, entitled "Screw Propellers," a chart is given which ties together the following elements in propeller design: Apparent slip; indicated thrust per square inch of disc area; ratio of projected area to disc area; propulsive coefficient and block coefficient of fineness of the hull.

From this list there is one element missing, and it is an element of considerable importance in propeller design, viz: tip speed. In order to take this missing element into consideration the chart shown on Plate 5 has been repeated in an extended form in this paper (Plate 8).

The changes made in this chart from that on Plate 5 are as follows: The propulsive-coefficient curves have been raised sufficiently to bring them above and clear of the apparent slip curves, and have been extended to the right to give estimated values of this coefficient at low values of indicated thrust per square inch of disc area; the apparent slip curves for projected-area ratios of .32 and .54 have been plotted for these same low values of indicated thrust, while those for a projected-area ratio of .54 at high values of thrust have been eliminated; in addition, below the slip curves have been plotted curves of tip speed for several values of projected-area ratios, using only such propellers as approximated very closely in the forms of projected areas to the Bureau Standard forms

and whose performances checked up very closely with the chart.

Cross curves of tip speeds at critical thrusts and at thrusts for maximum propulsive coefficients are shown with these other curves of tip speeds.

The method of design in which these curves of tip speeds are used is shown by the following example, in which revolutions, R, are replaced by tip speed (T. S.) divided by $\pi \times$ diameter (π D), and the two equations for pitch (P) and diameter are derived as follows:

Let T.S. = tip speed, in feet per minute;
s = percentage slip, from chart;
P.C. = propulsive coefficient, from chart;
I.H.P. = indicated horsepower = S.H.P. $\div$ .92;
E.H.P. = effective horsepower = I.H.P. $\times$ P.C.;
S = speed corresponding to E.H.P. and is taken from the model-tank curves, the hull of the model when towed being fitted with all appendages;
P = pitch of propeller, in feet;
D = diameter of propeller, in feet;
R = revolutions per minute of propeller $= \frac{\text{T.S.}}{\pi\,\text{D}}$;

$$\text{P} \times \text{R} = \text{P} \times \frac{\text{T.S.}}{\pi\text{D}}.$$

Speed of ship per minute = 101.33 $\times$ S.

$$= (1 - s)\ \text{P} \times \text{R} = \frac{\text{P} \times \text{T.S.} \times (1 - s)}{\pi\text{D}}.$$

d = apparent slip of propeller in feet per minute.

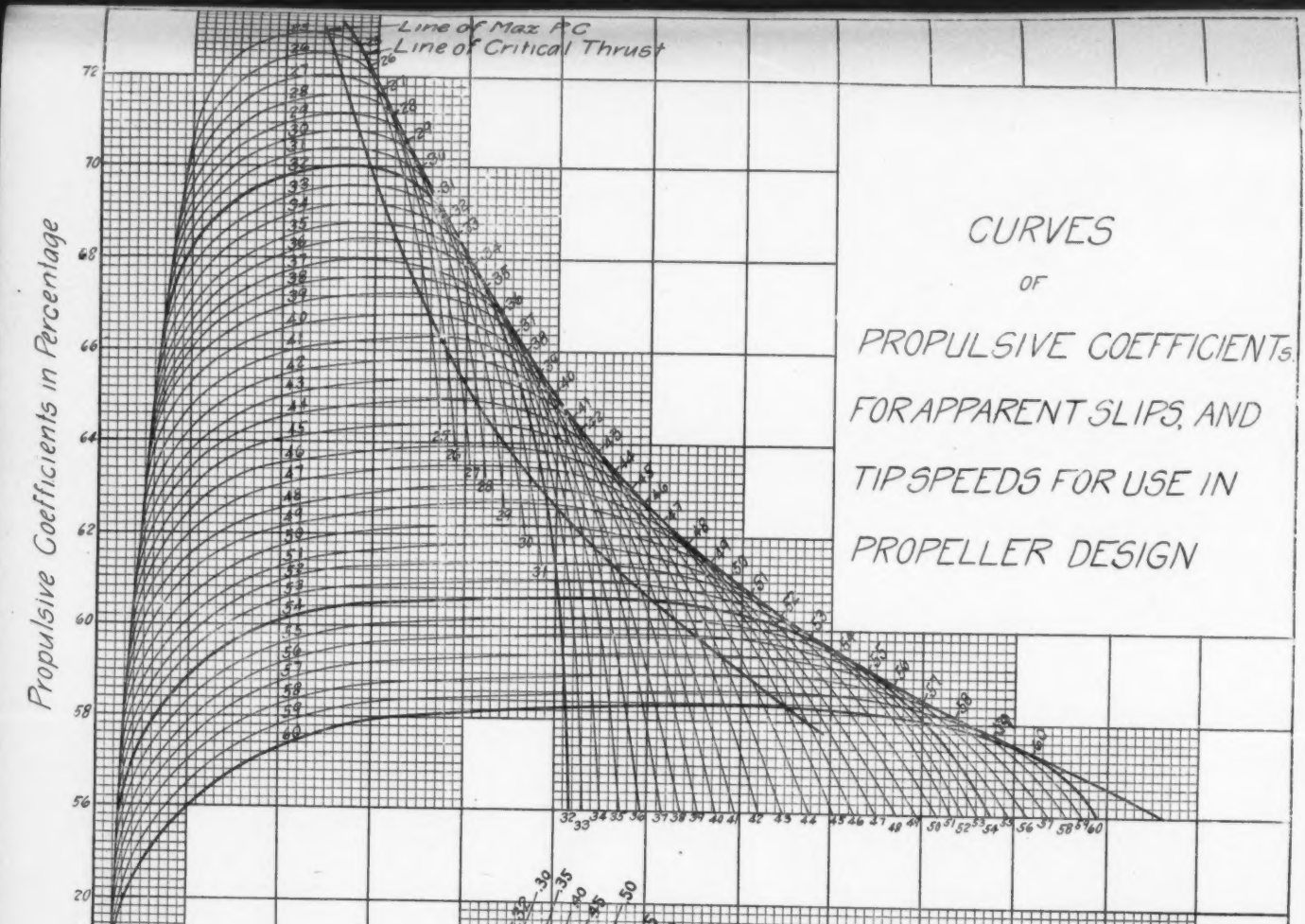
$$\text{d} = \text{P} \times \text{R} - \text{P} \times \text{R} \times (1 - s) = \text{P} \times \text{R} \times s = \frac{\text{P} \times \text{T.S.} \times s}{\pi\text{D}}.$$

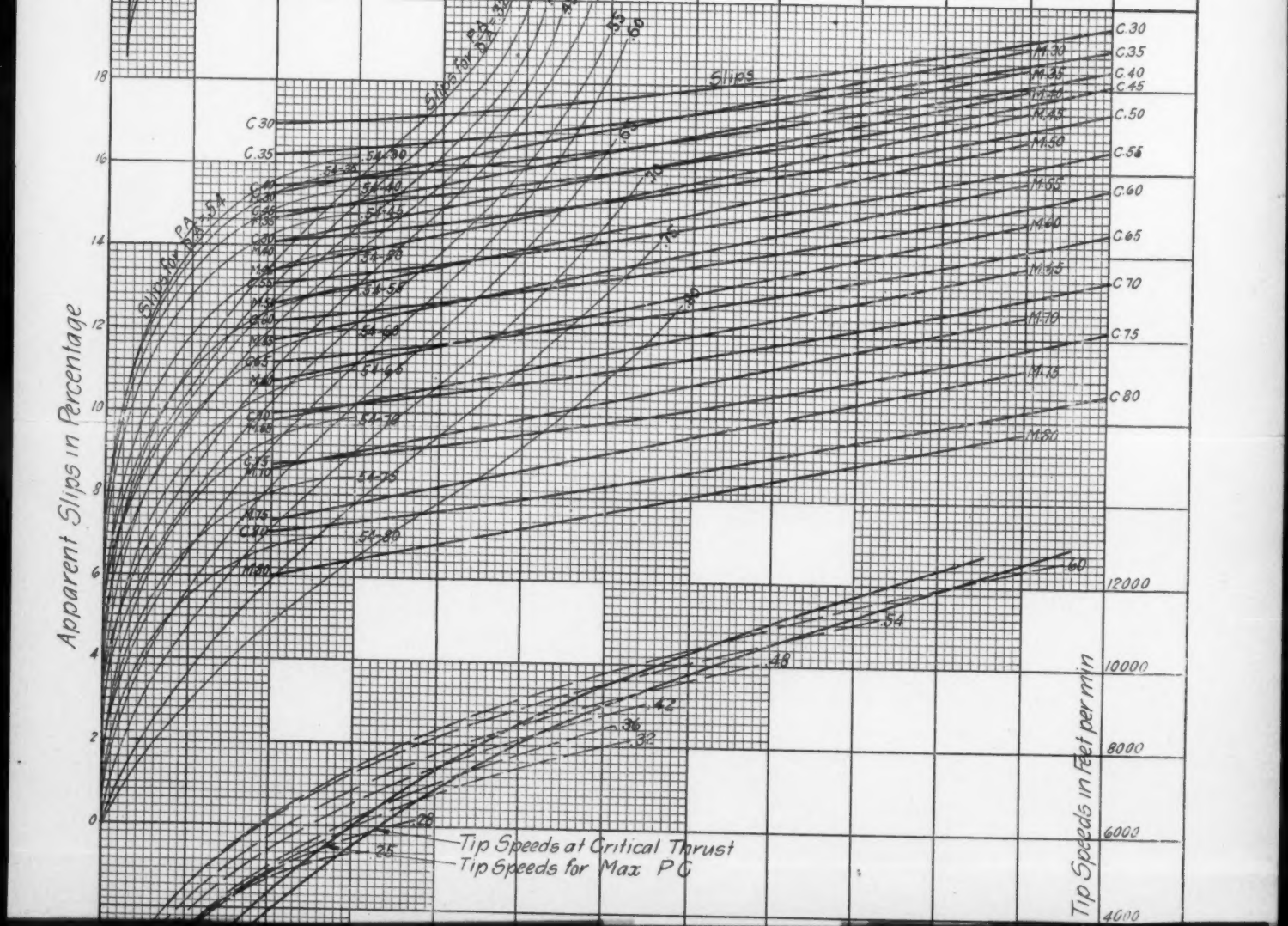
$$\therefore\ \text{P} = \frac{\text{d} \times \pi\,\text{D}}{s \times \text{T.S.}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

Total indicated thrust = total I.T. =

$$\text{I.H.P.} \times 33{,}000 \div \text{P} \times \text{R} = \text{I.H.P.} \times 33{,}000 \div \text{P} \times \frac{\text{T.S.}}{\pi\,\text{D}}.$$

Substituting in this equation the value of P from equation


Line of Max A.C.
Line of Critical Thrust
Propulsive Coefficients in Percentage
CURVES
OF
PROPULSIVE COEFFICIENTS
FOR APPARENT SLIPS, AND
TIP SPEEDS FOR USE IN
PROPELLER DESIGN


Apparent Slips in Percentage
Tip Speeds in Feet per min
Slips
Tip Speeds at Critical Thrust
Tip Speeds for Max P C
C.30
C.35
C.40
C.45
C.50
C.55
C.60
C.65
C.70
C.75
C.80
M.30
M.35
M.40
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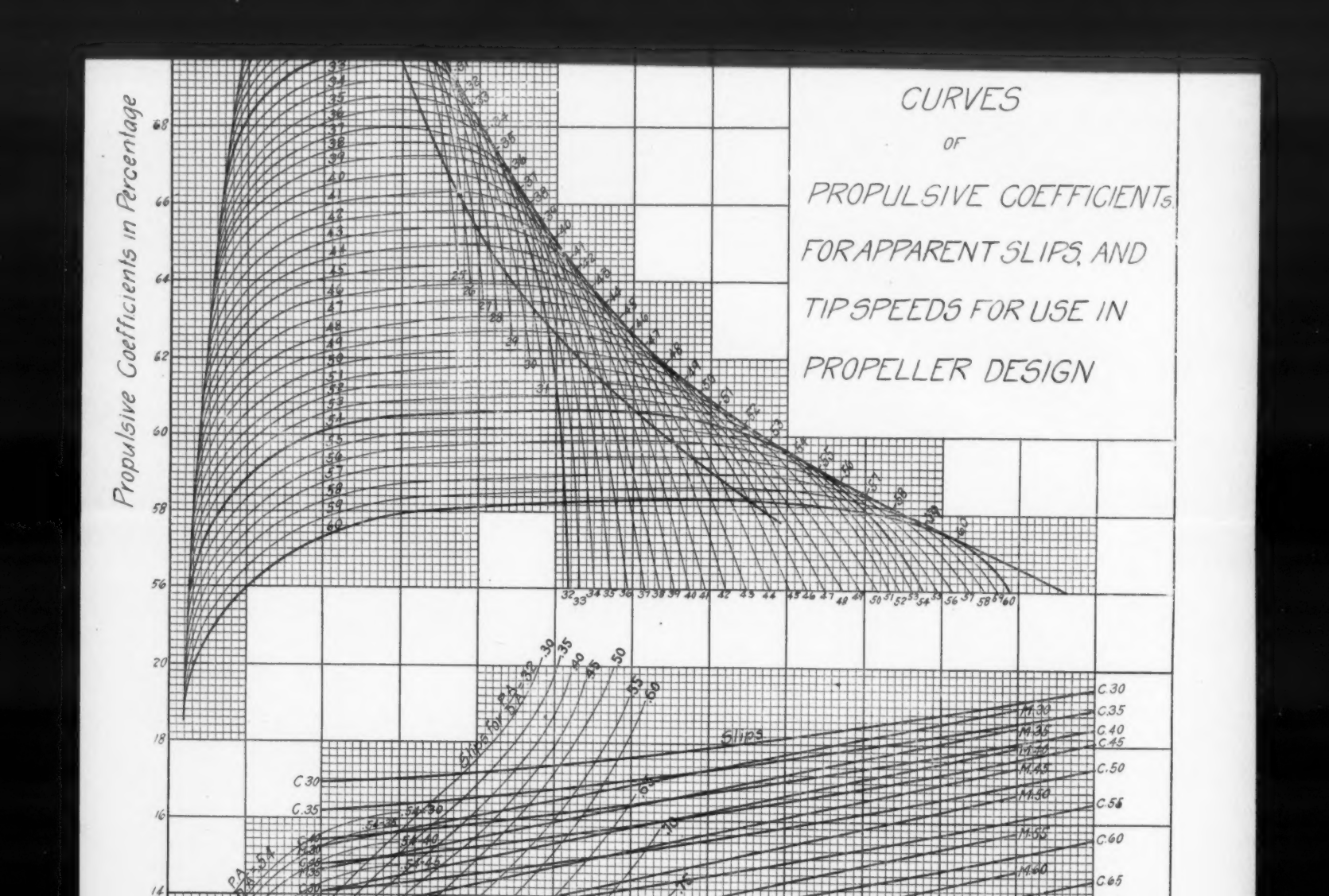
CURVES
OF
PROPULSIVE COEFFICIENTS
FOR APPARENT SLIPS AND
TIP SPEEDS FOR USE IN
PROPELLER DESIGN
Propulsive Coefficients in Percentage
Slips
Slips (%)
C.30
C.35
C.40
C.45
C.50
C.55
C.60
C.65
M.30
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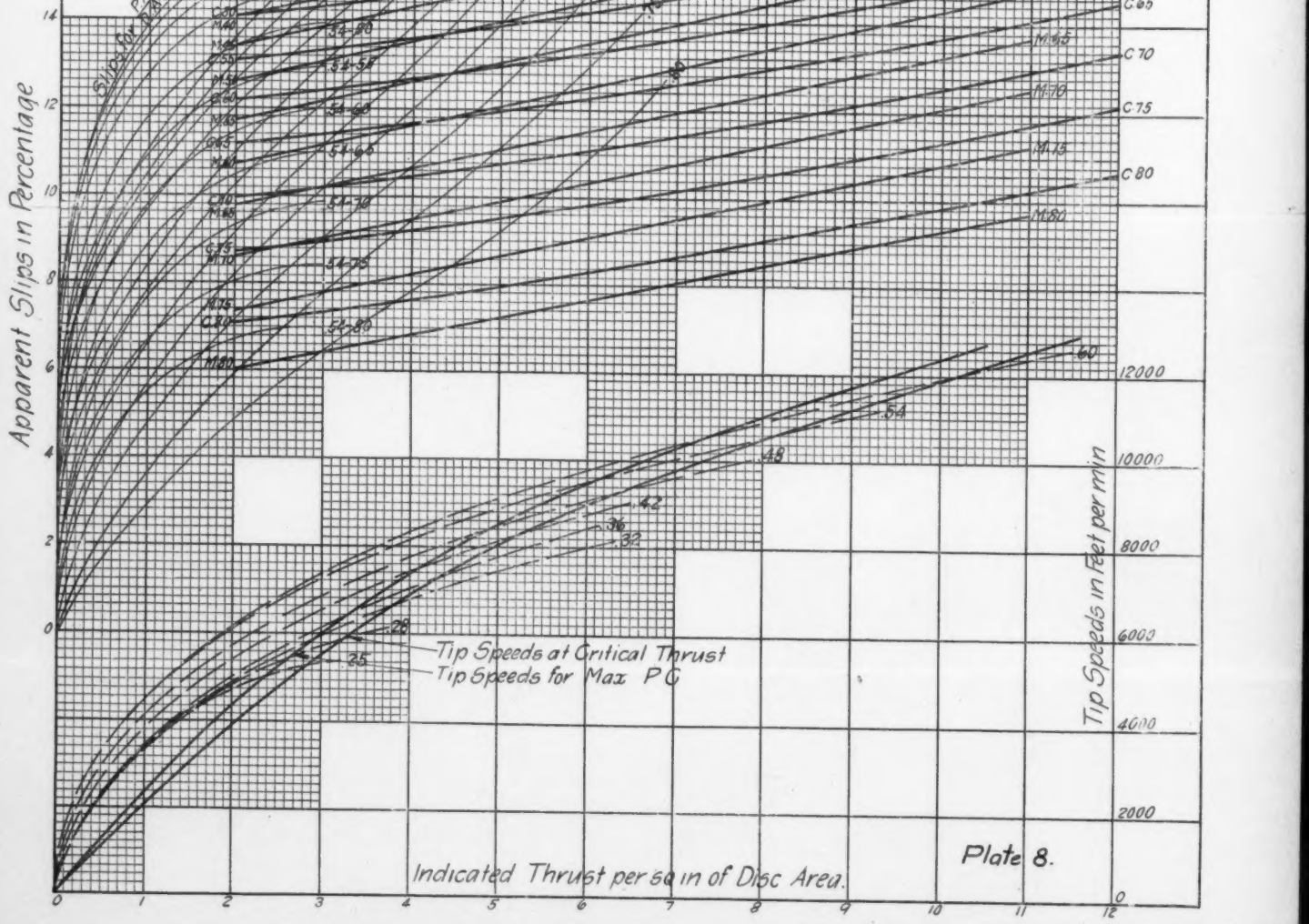
M.55
M.60


Apparent Slips in Percentage
Tip Speeds in Feet per min
Indicated Thrust per sq in of Disc Area.
Tip Speeds at Critical Thrust
Tip Speeds for Max P C
Plate 8.
C 65
C 70
C 75
C 80
M 65
M 70
M 75
M 80
.60
.54
.48
.42
.36
.32
.28
.25
0
2
4
6
8
10
12
14
2000
4000
6000
8000
10000
12000
0
1
2
3
4
5
6
7
8
9
10
11
12

(1), we have

$$\text{Total indicated thrust} = \text{I.H.P.} \times 33{,}000 \div \frac{d \times \pi D}{s \times \text{T.S.}} \times \frac{\text{T. S.}}{\pi D} = \frac{\text{I.HP.} \times 33{,}000 \times s}{d}.$$

Disc area of propeller, in square inches, $= \frac{1}{4}\pi D^2 \times 144$.

I.T. = indicated thrust per square inch of disc area corresponding to the T.S. and projected-area ratio used, and is taken from the chart.

$$\text{I.T.} = \frac{\text{Total indicated thrust}}{\text{Disc area in square inches}}.$$

$$\therefore \text{I.T.} = \frac{\text{I.H.P.} \times 33{,}000 \times s}{d} \div \tfrac{1}{4}\pi D^2 \times 144 = \frac{4 \times \text{I.H.P.} \times 33{,}000 \times s}{d \times \pi \times D^2 \times 144}.$$

$$\therefore D^2 = \frac{4 \times \text{I.H.P.} \times 33{,}000 \times s}{d \times \pi \times 144 \times \text{I.T.}} = \frac{291.78 \times \text{I.H.P.} \times s}{d \times \text{I.T.}}. \quad . \quad . \quad (2)$$

APPLICATION OF EQUATIONS (1) AND (2).

1. High Power, Moderate Number of Revolutions, Limited Diameter.

Block coefficient of fineness of vessel = .60;
Designed I.H.P. of engines = 25,000;
Number of propellers = 2;
I.H.P. per propeller = 12,500;
Revolutions = 122;
Maximum allowable diameter of propeller = 18 feet 3 inches;
Maximum tip speed for this diameter =

$$18.25 \times \pi \times 122 = 6{,}995 \text{ feet.}$$

Beginning with a projected-area ratio of .342, which corresponds to this tip speed on the curve of tip speeds at critical thrusts, maintaining this tip speed and using various P.A. ratios on each side of .342, we have, by taking the I.T.s, projected-area ratios, apparent slips and propulsive coefficients corresponding to this tip speed, as plotted on the curve of tip

speeds for the various P.A. ratios, from the chart, and tabulating them, we have

Tip speed allowed, in feet per minute = T. S.	P. A. ÷ D. A. from chart.	I. T. per square inch disc from chart.	Propulsive coefficient = P. C.	Designed revolutions.	Apparent slip = s from chart.	I.H.P. on one propeller.	E.H.P. on one propeller = I.H.P. × P. C.	Speed = S from tank curves.	S × 101.33 = distance moved per minute by ship.	P × R = S × 101.33 ÷ (1 − s).	d = P × R − S × 101.33.	$D=\sqrt[4]{\frac{33{,}000 \times \text{I.H.P.} \times s}{d \times \pi \times 144 \times \text{I.T.}}}$	$P=\frac{d \times \pi \times D}{s \times \text{T.S.}}$	Revs. $=\frac{\text{T. S.}}{\pi D}$
6,995	.355	4.0	.6765	122	.123	12,500	8,446	21.36	2,164.41	2,467.97	303.56	19.2	21.28	less than 122
6,995	3.5	4.05	.678	122	.125	12,500	8,475	21.38	2,166.44	2,475.93	309.49	19.05	21 18	less than 122
6,995	.342	4.1	.678	122	.127	12,500	8,725	21.38	2,166 44	2,481.60	315.16	18.912	21.08	less than 122
6,995	.33	4.2	.6765	122	.134	12,500	8,446	21.36	2,164.41	2,499.32	334.91	18.619	20.90	less than 122
6,995	.32	4.3	.670	122	.136	12,500	8,375	21.32	2,160.36	2,500.41	340.05	18 397	20.66	less than 122
6,995	.315	4.4	.661	122	.142	12,500	8,326	21.30	2,158.33	2,515.54	357.21	17.925	20 25	more than 122

The revolutions for equal tip speeds will vary inversely as the diameters; therefore all but the last propeller of the table will turn up too slowly, while the last one will be slightly in excess of the desired revolutions.

To obtain the propeller which will meet all requirements we lay down curves of pitch-diameter and of $\frac{\text{P.A.}}{\text{D.A.}}$-diameter, from which we can obtain directly the pitch and projected-area ratio required with a diameter of 18 feet 3 inches. This propeller will not be as efficient at high powers as those higher in the table, but its efficiency through the lower ranges of power will greatly exceed those of larger area and diameter. (See plate 9.)

Rule 1. Should the propeller obtained from the chart by using the P.A. ratio, slip and I.T. per square inch of disc area corresponding to the allowed tip speed, taken from the curve of tip speeds for critical thrust, have a greater diameter than can be fitted, we must use greater I.T.s, the P.A. ratios, propulsive coefficients and slips, corresponding to these new I.T.s,

the tip speeds for these additional screws being the same as that used before.

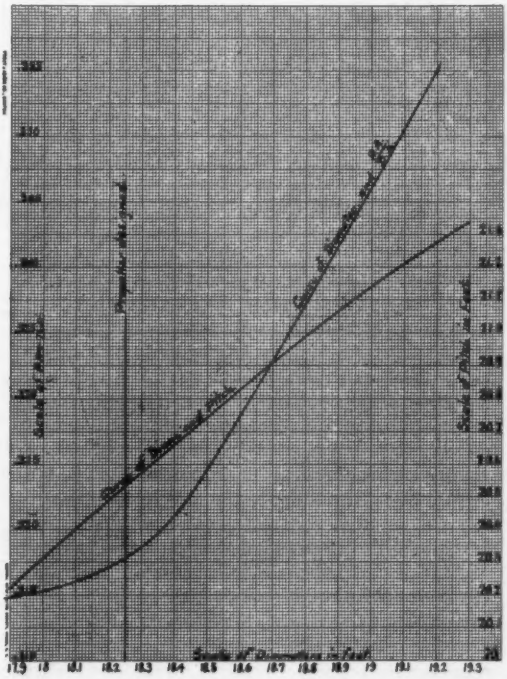


Plate 9.

Rule 2. Should the propeller obtained from the chart have a less diameter than that giving the necessary revolutions and tip speed, lower I.T.s per square inch of disc corresponding to this same tip speed, with the greater values of P.A. ratios, and with the corresponding slips and propulsive coefficients must be used.

2. Moderate Power, High Number of Revolutions, Limited $\frac{P}{D}$

In designing propellers for use with turbines, on account of the low pitch used, in order to avoid too low a pitch angle

at the tip it is usual to fix upon a limiting lower value of this pitch ratio. In general practice with Parsons' turbines this limit is usually fixed at .9. When such a limit is fixed, equation (1) becomes

$$P=\frac{d\times\pi\times D}{s\times T.S.}=.9D. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1_M)$$

$$\therefore \frac{\pi d}{s\times T.S.}=.9\,;\ \therefore T.S.=\frac{\pi d}{.9\times s}\,;$$

and equation (2),

$$D^2=\frac{4\times 33{,}000\times I.H.P.\times s}{d\times\pi\times 144\times I.T.}=\frac{33{,}000\times I.H.P.}{.9\times T.S.\times 36\times I.T.}=$$

$$\frac{11{,}000\times I.H.P.}{10.8\times T.S.\times I.T.}=\frac{1{,}018.52\times I.H.P.}{T.S.\times I.T.} \quad . \quad (2_M)$$

In order to insure the efficiency of the turbine the revolutions, R, are fixed for the designed power; therefore, conversely, having fixed the number of revolutions of the propeller to insure proper efficiency at the desired speed, the diameter of the turbines for proper vane speed can be obtained.

To apply these modified formulas, (1_M) and (2_M), proceed as in the following problem:

Block coefficient of fineness=.45;

Speed=28 knots;

$$\text{Shaft horsepower}=10{,}000\,;\ \therefore I.H.P.=\frac{10{,}000}{.92}=10{,}870\,;$$

Number of shafts=3;

I.H.P. per propeller=3,624.

Required Revolutions per Minute, Diameter and Pitch of Propellers. Ratio of Pitch to Diameter to be .9.

Ratio of Pitch to Diameter to be .9.

P.A.	Ratio=.56,	P.C.=58.9,	E.H.P.=6,402,	Speed=28.02
"	" =.57,	P.C.=58.6,	E.H.P.=6,370,	" =28.01
"	" =.58,	P.C.=58.2,	E.H.P.=6,326,	" =28.00
"	" =.59,	P.C.=57.8,	E.H.P.=6,283,	" =27.99
"	" =.60,	P.C.=57.4,	E.H.P.=6,238,	" =27.98

P.A./D.A.	P.C.	I.H.P.	Speed.	Slip.	101.33× speed.	P×R=101.33S ÷(1−s).	d.	I.T.	T.S. and πd÷.9s.
.56	.589	3,624	28.02	.1679	2,839.27	3,412.17	572.90	8.7	11,000 (C) 11,910 (I_M)
.57	.586	3,624	28.01	.1688	2,838.25	3,414.64	576.39	9.0	11,200 (C) 11,919 (I_M)
.58	.582	3,624	28.00	.1700	1,837.26	3,418.39	581.13	9.37	11,450 (C) 11,933 (I_M)
.59	.578	3,624	27.99	.1715	2,836.25	3,423.36	587.11	9.79	11,700 (C) 11,949 (I_M)
.60	.574	3,624	27.98	.1731	2,835.23	3,428.75	593.52	10.2	12,000 (C) 11,969 (I_M)

Laying down the tip speeds from chart and as derived by equation (I_M), with $\frac{P.A.}{D.A.}$ as abscissas, as shown in Plate 10, we obtain two curves which intersect. The propeller designed with the projected-area ratio at the point of intersection will be the required propeller. It is as follows:

$\frac{P.A.}{D.A.}$,	0.5990
P.C.,	0.5761
Tip speed,	11,966.0
Diameter, feet,	5.513
Pitch, feet,	4.9617
Slip,	17.3
Speed,	27.986
I.H.P.,	3,624.0
I.T.,	10.15
Revolutions,	690.8

Very often, particularly with slow, heavy ships with a low number of revolutions of the propeller, say below 100 per minute, the tip speed is below 5,000 feet. In this case, to hold down the revolutions of the engine we must, in using the chart of design, take higher projected-area ratios than that corresponding to the tip speed measured on the curve of tip speed for critical thrusts; that is, we must take the I.T.s and slips corresponding to the tip speed as measured on the curve of tip speeds for each projected area used, thus:

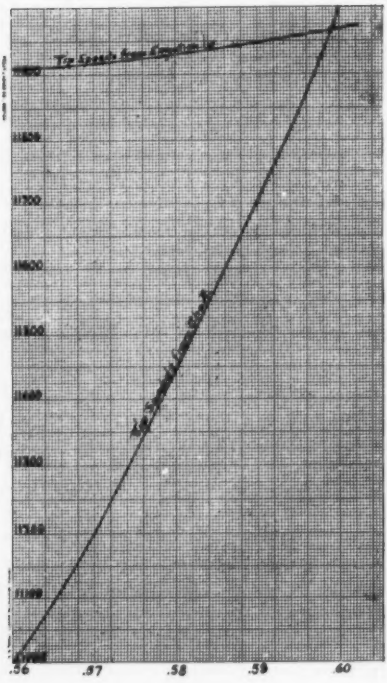


Plate 10.

Low Revolutions, Low Power and Low Tip Speed.

Block coefficient of fineness of hull=.8;
I.H.P.=4,000 on two propellers=2,000 for each propeller;
Maximum diameter allowed=14 feet 9 inches;
Revolutions of propellers for this power=95;
Tip speed per minute=4,402 feet.

In the part of the chart here used, as the tip speed increases, the efficiency rises until the curve of maximum efficiency is crossed, so that we really should, if possible, design our propeller for the tip speed corresponding to this point of maximum efficiency, this higher tip speed being obtained by increase in diameter and not in revolutions. This is not always possible, however, as we are limited in our diameter.

P.A. D.A.	P.C.	I.H.P.	E.H.P.	Speed (S).	Slip (s).	101.33 ×S.	P×R.	d.	D.	P.	T.S.	I.T.	Proj. A.
									feet.	*feet.*			
.25	.727	2,000	1,454	13.30	.043	1,347.69	1,408.22	60.53	15.85	15.92	4,402	1.65	49.33
.27	.715	2,000	1,430	13.25	.043	1,342.62	1,402.95	60.33	15.97	15.99	4,402	1.63	54.11
.29	.706	2,000	1,412	13.20	.043	1,337.56	1,397.66	60.10	16.10	16.06	4,402	1.61	59.07
.33	.689	2,000	1,378	13.13	.043	1,330.46	1,390.24	59.78	16.40	16.27	4,402	1.56	69.74
.36	.676	2,000	1,342	13.11	.043	1,328.44	1,388.13	59.69	16.74	16.58	4,402	1.50	79 24

On examination of the column giving D it is found that all of the above propellers are greater in diameter than the required propeller, and the revolutions for the same power will be lower in inverse ratio of the diameters. The I.T. per square inch of disc area of the required propeller of 14 feet 9 inches diameter will vary inversely as the diameters.

The projected area of the required propeller will be sufficient to give the same indicated thrust per square inch of projected area as the chart propeller had from which it is deduced.

The total thrust must equal that of the corresponding chart propeller. The slip will vary inversely as the cubes of the diameters, as it will vary directly as the thrust per square inch of disc area and inversely as the cross-sectional area of the column of water in the propeller race.

The pitch will vary inversely as the revolutions. Diameter, 14.75 feet.

$$P_1=\frac{1,408.22}{95}=14.823 \text{ feet.} \qquad Slip_1=\frac{.043\times\overline{15.85}^3}{14.75^3}=.0534$$

$$P_2=\frac{1,402.95}{95}=14.768 \qquad Slip_2=\frac{.043\times\overline{15.97}^3}{14.75^3}=.0546$$

$$P_3=\frac{1,397.66}{95}=14.712 \qquad Slip_3=\frac{.043\times\overline{16.10}^3}{14.75^3}=.0559$$

$$P_4=\frac{1,390.24}{95}=14.634 \qquad Slip_4=\frac{.043\times\overline{16.40}^3}{14.75^3}=.0591$$

$$P_5=\frac{1,388.13}{95}=14.612 \qquad Slip_5=\frac{.043\times\overline{16.74}^3}{14.75^3}=.0629$$

$$\text{Speed}_1 = S_1 = \frac{1{,}408.22 \times (1 - .0534)}{101.33} = 13.16;$$

$$S_2 = \frac{1{,}402.95 \times (1 - .0546)}{101.33} = 13.09;$$

$$S_3 = \frac{1{,}397.66 \times (1 - .0559)}{101.33} = 13.02;$$

$$S_4 = \frac{1{,}390.24 \times (1 - .0591)}{101.33} = 12.91;$$

$$S_5 = \frac{1{,}388.13 \times (1 - .0629)}{101.33} = 12.84.$$

As the I.T.s per square foot of projected areas of the deduced propellers must equal that of their corresponding chart propellers, and as the total I.T.s of the deduced and of the chart propellers are equal, the projected areas must be equal, and the projected area ratio of the new screw will be to that of the chart one, inversely as the squares of the diameters.

$$\therefore \frac{\text{P. A.}_1}{\text{D.A.}} = .25 \times \frac{\overline{15.85}^2}{14.75^2} = .289$$

$$\frac{\text{P. A.}}{\text{D.A.}^2} = .27 \times \frac{\overline{15.97}^2}{14.75^2} = .317$$

$$\frac{\text{P. A.}_3}{\text{D.A.}} = .29 \times \frac{\overline{16.1}^2}{14.75^2} = .346$$

$$\frac{\text{P. A.}}{\text{D.A.}^4} = .33 \times \frac{\overline{16.4}^2}{14.75^2} = .408$$

$$\frac{\text{P. A.}}{\text{D.A.}^5} = .36 \times \frac{\overline{16.74}^2}{14.75^2} = .464$$

These points of pitch, slip and projected-area ratio of the 14.75-foot-diameter propellers, and the resultant speed of vessels can now be plotted as curves on Plate 11, and from these

curves that propeller picked out which, in our opinion, will give the best general satisfaction. In arriving at this opinion, we must take into consideration the difference between the loading of the vessel in service and that on trial, and should deliberately sacrifice some efficiency on trial to gain in efficiency with the heavier load.

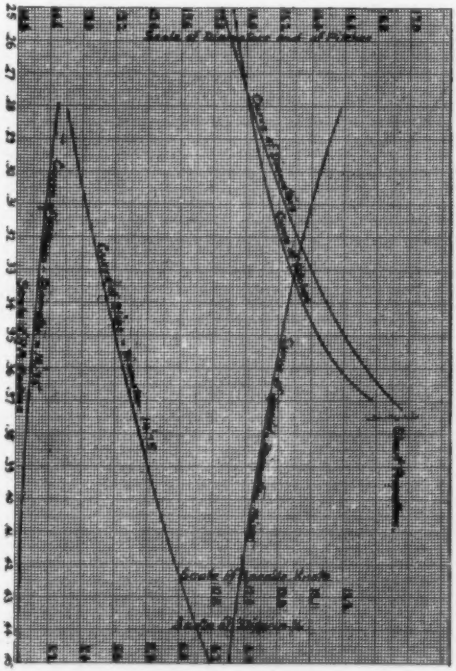


Plate 11.

This last method of computing a propeller should also be used for launch propellers where, though the revolutions are comparatively high, the diameters of the propellers are small and the tip speeds low.

Analysis of a propeller submitted for a certain duty.

To analyze the expected performance of a propeller we have given us the pitch, diameter, projected area and designed revolutions for the designed power. In addition, we have at our disposal the model-tank curves of E.H.P. for the vessel to which the propeller is to be applied.

Let us assume the following characteristics:

Block coefficient of fineness of vessel = .60;
Diameter of propeller = 18 feet 3 inches;
Pitch of propeller = 19 feet 9 inches;
$\frac{\text{P.A.}}{\text{D.A.}}$ of propeller = .328;
Designed revolutions at designed power = 125;
Total designed power = 25,000 I.H.P.;
Power applied to one propeller = 12,500;
Tip speed of propeller = $125 \times \pi \times 18.25 = 7{,}167$ feet per minute.

Taking data from the chart and the corresponding speeds from the model-tank curves, we have—

$\frac{\text{P.A.}}{\text{D.A.}}$	P.C.	I.T.	Slip (s).	I.H.P.	E.H.P	Speed (S).	101.33 × (s).	P. × R.	d.	D.	P.	Proj. area.
										feet.	*feet.*	
.32	.661	4.55	.141	12,500	8,263	21.27	2,151.99	2,505.23	353.24	17.888	19.644	80.42
.33	.665	4.45	.137	12,500	8,313	21.30	2,155.03	2,497.14	342.11	18.117	19.83	85.07
.34	.669	4.35	.133	12,500	8,363	21.33	2,158.07	2,489.12	331.05	18.354	20.025	89.95
.35	.673	4.25	.129	12,500	8,413	21.36	2,161.11	2,481.18	320.07	18.578	20.206	94.88
.37	.674	4.10	.123	12,500	8,517	21.40	2,168.46	2,472.60	304.14	18.967	20.558	104.55

On Plate 12 are plotted curves of pitches—diameters, projected-area ratios—diameters, projected areas—diameters, and slips-diameters.

From these curves we see that a propeller to make 125 revolutions with 12,500 I.H.P., under the conditions of the problem will have the following dimensions:

Diameter, 18.25 feet; pitch, 19.94 feet; P.A. ratio, .3355; P.A., 87.7 square feet.

From the same curves, with a pitch of 19.75 feet, the diameter will be 18.02 feet; P.A. ratio, .3256; P.A., 83 square feet; the revolutions, 126.45. While with a projected area equal to that of the propeller submitted, namely, 85.8 square feet, from the same curves, the pitch will be 19.86 feet; P.A. ratio, .3315; diameter, 18.155 feet; revolutions, 125.7.

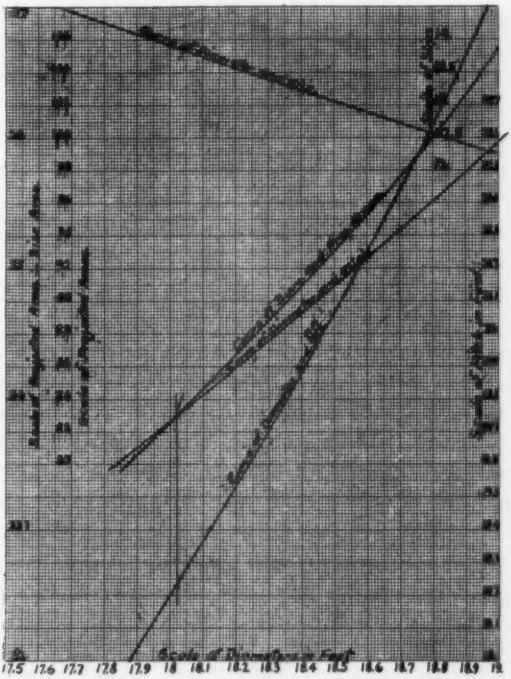


Plate 12.

The propeller submitted, having less pitch and less surface than the 18.25-foot-diameter propeller, from Plate 12, will turn up slightly higher. Having less pitch and slightly greater diameter than the 18.155-foot-diameter propeller on Plate 12, these differences may nearly offset each other, and we would expect about the same number of revolutions as

with this latter screw, particularly as the propeller under discussion has more diameter and surface than the 18.02-foot-diameter propeller having the same pitch.

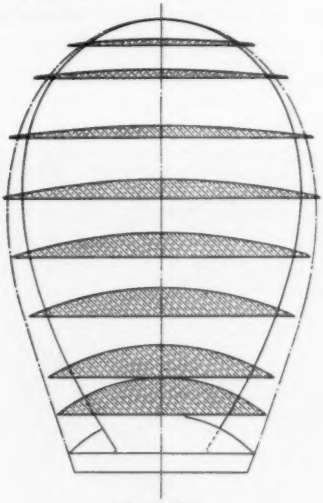


Plate 13.

Finally, having the same pitch, but greater diameter and surface than the 18.02-foot propeller from Plate 12, the revolutions will be slightly lower than those of this latter propeller, and the percentage slip will be slightly decreased, the decrease in slip possibly offsetting the decrease in revolutions, so that we should expect this propeller to give us approximately the same results as those given by the smaller chart screw of 18.02 feet in diameter.

BLADE SECTION AND CONDITION OF BLADE SURFACE ON WHICH THE FOREGOING WORK HAS BEEN BASED.

Blade Sections.—On Plate 13 are shown the forms of blade sections of the propellers from which the chart of design has been obtained. Any radical departure from this form of section will cause a propeller to differ in the tip speed obtained for any value of I.T. per square inch of disc area from the tip speeds for those thrusts as given on the chart.

The amount of difference in these tip speeds will depend upon the degree of variance from the standard forms of the form of section and the form of the projected area thus:

Blades having the metal of the thickness, distributed more or less evenly on each side of the nominal pitch surface will have much higher tip speeds than those of the standard blades. Blades having blunt entrances and leading edges will turn up slower.

Blades broader at the tip than the standard form will turn slower and those narrower at the tip than the standard will turn faster than the tip speeds given by Plate 8.

In only one case is it advantageous to depart from the standard section, and that is when through existing conditions it becomes necessary, in order to insure sufficient blade strength, to make the blade near the root very thick as compared with its width. In such a case the leading edge of the screw should be thrown slightly back from the nominal pitch surface in order to fine the entrance. This throw-back, however, should gradually melt away as we move out towards the tip, and when about one-half the radius out should have entirely disappeared, so that from this point out the nominal true-pitch surface will extend the full width of the blade.

Condition of Blade Surface.—In all the cases used in the construction of the chart of design the blades were machined to designed pitch and were set at that pitch. The surfaces, both driving face and back, were highly polished. These points have high value at all speeds of rotation, but are especially valuable at very high speeds. Any blade roughness, or considerable variation in pitch setting of the blades from the designed pitch may seriously affect the performance of the propeller, while the blade roughness will surely slow it down to lower tip speeds than given by the chart.

APPLICATION OF THE CHART OF DESIGN.

The chart may be used in designing screws for vessels fitted with any number of shafts, and confidence be felt in the resultant screws, provided the shafts are widely enough sepa-

rated to insure against any propeller working in the wake of one forward of it, and provided that in the case of vessels having a shaft in the middle line the dead wood be cut away sufficiently to insure that the middle screw works in nearly as good water as the wing screws.

If this is not done the resultant propulsive coefficient will be considerably, say roughly from 5 to 10 per cent., lower than that given by the chart, as the conditions under which a propeller working in the wake of and close to the stern post are not covered by the chart.

For such propellers the wake in which they work is much heavier than that to which the wing screws are subjected, and this heavier wake would necessitate larger diameter, more surface and lower pitch for the middle screw than for its companions. To compute the propellers accurately the wing screws should be obtained from the existing chart, while for the center screw a separate chart should be used prepared from the performances of propellers on single-screw ships. The writer, unfortunately, has not at hand sufficient reliable data to prepare such a chart, and must therefore use the existing chart, allowing for an expected falling off in efficiency where the adverse conditions for the middle screw exist.

THE CRUISE OF THE SUBMARINE TORPEDO BOAT *SALMON* FROM QUINCY, MASS., TO HAMILTON, BERMUDA, AND RETURN, JULY 5–17, 1910.

By Lieutenant D. A. Weaver, U. S. N., Member, Commanding Officer U. S. S. "Salmon."

At 2:00 P. M., July 5, 1910, the submarine torpedo boat *Salmon* sailed from Quincy, Mass., for a cruise to Hamilton, Bermuda, and return. This cruise was made before delivery of the vessel to the Government, under the direction of the contractors, the Electric Boat Co., but with the authority of the Secretary of the Navy.

The cruise is of interest because it is the first attempt by any submarine of the type built by the Electric Boat Company for the U. S. Government to make an extended deep-sea cruise under its own power.

The *Salmon* is of the Holland type, "modified spindle form," as opposed to the "ship-shape" forms of the Lake, Laurenti, Laubeuf, and Krupp types. All submarines in commission in the U. S. Navy are of the Holland modified spindle form type.

Without entering into a discussion of the relative merits of the various types of submarines it may be stated that the designers of the *Salmon* type claim (1) superior submerged speed for a given horsepower, and (2) superior structural strength for deep submergence.

The advocates of the "ship-shape" form claim that the superior submerged speed and structural strength of the "spindle form" are obtained at the expense of habitability and sea-going qualities. Keeping this statement in mind, the

reader may understand the request of the Electric Boat Company to make a deep-sea cruise (builder's trial) with the *Salmon* to obtain information as to habitability for long cruises at sea and the cruising capabilities of the *Salmon.*

The following are the approximate principal dimensions of the *Salmon:*

Length of spindle, feet and inches	128–04
Length over all, feet and inches	134–10
Breadth, molded, feet and inches	13–09
Breadth, extreme, feet and inches	13–10⅝
Total displacement submerged, tons	337
Depth under side of duct keel to top of superstructure, feet and inches	15–02⅝

The vessel is equipped with two 6-cylinder, 4-cycle, 300-H.P., gasoline engines, 12-inch diameter, and 12-inch stroke, with jump-spark ignition from storage batteries.

The crew of the *Salmon* for this voyage was composed of thirteen men from the shops of the builders. In addition there were on board Captain A. Cuevas, Chilean Navy, Mr. G. C. Davison, Vice-President of the Electric Boat Company, and until recently a Lieutenant Commander, U. S. Navy, Lieutenant D. A. Weaver, U. S. Navy, Assistant Naval Constructor D. R. Battles, U. S. Navy, and four enlisted men of the Navy, a total of twenty-one men on board for the cruise. Mr. G. C. Davison acted as navigator. The *Salmon* is fitted with fourteen berths, only six of which were used on the cruise, the others having been removed to make room for a table and stowage of spare parts and provisions. The men not assigned to berths slept on air mattresses on the battery decks.

DATA AT BEGINNING OF CRUISE.

Gasoline in fuel tanks, gallons	6,000
Lubricating oil on board, gallons	385
Fresh water on board, gallons	1,085

Rations 21 men—7 days, pounds	1,000
Draught, forward, feet and inches	12–3
Draught, aft, feet and inches	12–3
Water in ballast tanks	None

There being no torpedoes on board, the four torpedo tubes were filled with fresh water. The total capacity of the fresh-water tanks on the *Salmon* is one hundred and eighty-five gallons, a quantity sufficient for drinking and cooking, but insufficient for washing purposes. Sixty-one five-gallon tins of lubricating oil were carried in the bilges, as the total capacity of the oil tanks is only eighty gallons. An ice chest 79 inches by 19½ inches by 42 inches was built in the space where the torpedoes are handled. The ice chest served also as a table, and was of great utility.

The run from Quincy to Provincetown was made on July 5th, under the port engine, the starboard engine having been disabled by a gasket blowing out under No. 3 cylinder head when starting. Arrived at Provincetown at 8:00 P. M., after swinging ship in Cape Cod Bay. The *Salmon* is equipped with four compasses: one 7-inch compensating compass in the after end of the cut-water (only visible from inside the boat), one 4-inch compensating compass in the conning tower, one 4-inch non-compensating movable compass for steering, that is clamped on brackets to the after periscope housing, and one 4-inch non-compensating compass fitted in the top of the forward periscope and can be seen only through the periscope, the rim of the compass marked in degrees 0 to 360, showing around the edge of the field of vision.

The insurance companies demanded that a convoy be taken on the cruise and the sea-going tug *Underwriter,* of the Boston Tow Boat Company, arrived at Provincetown at 10:30 P. M., July 5th. The *Underwriter* was instructed to follow the *Salmon,* the purpose being to have the *Salmon* rely on her own resources as to navigation.

SUMMARY OF DECK LOG.

July 6th.—At Provincetown, Mass. Got underway at 9:00 A. M.; speed, 10 knots, 400 revolutions both engines, and set a course for Hamilton, Bermuda.

Position at noon:

Latitude	41° 59′ N.
Longitude	69° 59′ W.
Distance made good since leaving Quincy	85 miles.
Gasoline used	380 gallons.
Weather since leaving Quincy	Fair.
Sea	Smooth.

On course S. 15° E. (Mag.).

Slowed down from 9:40 P. M., to 11:10 P. M., on account of thick fog.

July 7th.—On course S. 15° E. (Mag.). Fresh breezes from S.W. and sea becoming rough. At 2:00 A. M., stopped port engine due to gasket blowing out under No. 2 cylinder head.

Noon position:

Latitude	39° 15′ N.
Longitude	68° 16′ W.
Distance made good preceding 24 hours	183 miles.
Gasoline used during preceding 24 hours	1,100 gallons.

Moderate sea on starboard bow during greater part of day. Angle of pitch about 5°, angle of roll 10° to 15°. Majority of the crew were sea-sick, and for this reason no attempt was made to replace the gasket on No. 2 cylinder, port engine.

July 8th.—On course S. 15° E. (Mag.), starboard engine; moderate sea; fresh breezes from S.W. At 9:00 A. M., changed course to S. 6° E. (Mag.), to allow for wind and current.

Position at noon:
Latitude.....36° 50′ N.
Longitude.....66° 26′ W.
Distance made good preceding 24 hours.....180 miles.
Gasoline used during preceding 24 hours.....748 gallons.

About 1:30 P. M., two heavy seas struck the bridge, carrying away the bridge screens and shipping about half a ton of water down the conning-tower hatch.

Abandoned bridge and closed conning-tower hatch, steering from inside the conning tower. Resumed station on bridge at midnight.

July 9th.—On course S. 6° E. (Mag.), under starboard engine; moderate breezes from S.S.W.; moderate sea; weather warm.

Noon position:
Latitude.....34° 08′ N.
Longitude.....65° 34′ W.
Distance made good preceding 24 hours.....170 miles.
Gasoline used during preceding 24 hours.....748 gallons.

At noon changed course to S. 18° E. (Mag.).

July 10th.—On course S. 18° E. (Mag.), starboard engine; weather moderating. At 1:00 A. M., sighted Gibbs Hill Light, and at 5:30 A. M., when off Saint Davids Head, took on Pilot. Moored along side *Underwriter* at city docks, Hamilton, Bermuda, at 8:30 A. M.

Distance made good preceding 24 hours.....150 miles.
Gasoline used during preceding 24 hours.....640 gallons.
Total distance from Quincy, Mass.....760 miles.
Total gasoline used from Quincy, Mass.....3,616 gallons.
Total lubricating oil, from Quincy, Mass..... 95 gallons.

July 13th.—Sailed from Hamilton at 9:45 A. M. At 11:50 A. M., changed course to N. 9° W. (Mag.); moderate sea; weather pleasant.

Distance made good preceding 24 hours.....25 miles.
Gasoline used during preceding 24 hours.....170 gallons.

July 14th.—On course N. 9° W. (Mag.); fresh breezes from the W.S.W.; sea becoming rough.

Noon position:
Latitude.....35° 54′ N.
Longitude.....66° 20′ W.
Distance made good preceding 24 hours.....218 miles.
Gasoline used during preceding 24 hours.....1,104 gallons.

July 15th.—On course N. 9° W. (Mag.); weather moderating.

Noon position:
Latitude.....39° 07′ N.
Longitude.....68° 20′ W.
Distance made good preceding 24 hours.....212 miles.
Gasoline used during preceding 24 hours.....1,185 gallons.

At 2:30 P. M., changed course to N. 22° W. (Mag.). At 5:00 P. M., thick fog shut in, and at 6:45 took tow line from the tug *Underwriter,* as the fog signal of the *Salmon* (whistle blown by compressed air) is not loud and it was difficult to keep in company with the tug. At 11:00 P. M., picked up Nantucket Light Ship.

July 16th.—On course N. 49° E. (Mag.); thick fog; sea smooth. At 12:28 A. M., fog lifted and tow line was cast off. Started ahead on both engines; speed, 10½ knots. Soon after starting fog shut in, and at 2:00 A. M. again took tow line from tug.

Noon position:
Latitude.....41° 55′ N.
Longitude.....69° 38′ W.
Distance made good preceding 24 hours.....200 miles.

At 1:00 P. M. the fog lifted, and when an attempt was made to start the engines it was found that all of the gasoline, about 3,000 gallons, had been blown overboard due to the deceptive reading of a gauge. All of the fuel tanks are not fitted with independent gauges. At 3:46 P. M., cast off the line from the *Underwriter* and proceeded into Provincetown under electric motors for gasoline, the *Underwriter* proceeding to Boston. Took on 640 gallons of gasoline.

July 17th.—Got underway at 7:00 A. M., and proceeded to Quincy; speed, about 11½ knots. Arrived at Quincy at 11:00 A. M.

Total distance of cruise.....1,514 miles.
Total gasoline (estimated).....7,256 gallons.

SUMMARY OF ENGINE-ROOM LOG.

July 5th.—In getting underway the gasket in No. 3 starboard cylinder was blown. Replaced gasket at Provincetown, Mass. Average revolutions port engine, 465. (Full speed, 515 revolutions.)

July 6th.—Got underway at 9:00 A. M., 400 revolutions both engines. At 3:10 P. M., stopped port engine on account of fuel strainers being clogged with a shellac composition from the paint in the fuel tanks. The fuel tanks were painted with a mixture of shellac and Portland cement in alcohol, Portland cement being added to the shellac until it can just be worked with a brush. Started port engine at 3:20 P. M. At 6:58 P. M., No. 3 induction-valve arm on port engine stuck on rocker shaft. Stopped port engine. At the same time replaced broken spool on No. 1 upper exhaust rocker arm. Started port engine at 7:18 P. M. Average revolutions, starboard, 400.

July 7th.—At 2:00 A. M., found water in No. 2 port cylinder. Stopped engine and renewed gasket. Started port engine at 7:30 P. M., and after about a half hour's attempt to

make the engine run, water was found in cylinders and crank pits from a leak in the gasket from No. 5 cylinder.

Most of the crew were too sea-sick to work on cylinder heads and the motion of the boat made the renewing of the gaskets difficult. Average revolutions, starboard, 400.

July 8th.—Had trouble with salt water in the fuel during the morning watch, due to water backing into fuel tanks through outboard vent. Shifted to inboard vent and to the short suction in fuel tank.

On account of heavy seas, closed conning-tower hatch at 1:30 P. M. Took air for engines through ventilators. Ventilation of boat was good.

July 9th.—Renewed gasket on No. 5 cylinder head, port engine. Average revolutions, starboard, 400.

July 10th.—Arrived Hamilton at 8:30 A. M. Full charged storage battery with starboard engine. Battery in good condition.

July 11th.—Took off cylinder heads of Nos. 3, 4, 5 and 6 starboard engine, and Nos. 3, 4 and 6 port engine, and renewed gaskets. A carbon deposit was found on the cylinder heads 1/32 to 1/16 inch thick. Replaced two leaky nipples on water-jacket vent line.

July 12th.—Finished work on cylinder heads. Examined main bearing caps. Took on gasoline, water, ice and fresh provisions.

July 13th.—Sailed for Boston at 9:45 A. M. While starting the port engine the threads in link connecting No. 4 port lower exhaust valve stem to dash-pot stem stripped and put No. 4 cylinder out of commission. Ran port engine on five cylinders and drilled a hole through yoke and stem for a pin, but the pin soon sheared off. No. 1 port upper exhaust valve did not close properly and a spring was suspended above it attached to the valve stem.

July 14th.—The lower exhaust valve on port No. 5 cylinder failed to function. It was discovered later that a piece was broken out of the valve.

The steering motor short-circuiting switch burned out due to defective push button in portable. Used hand-steering gear. Average revolutions, both engines, 400.

July 15th.—Had trouble with the ignition system port engine, finally located in the distributor. Put in new sprocket. On tow line. Charged air with starboard compressor, 103 cubic feet from 1,000 pounds to 2,000 pounds. Time, about 3½ hours.

July 16th.—Tried to start the engines at 6:00 A. M., but found water in the fuel; the starboard engine ran nearly an hour when all of the gasoline gave out. Went into Provincetown under electric motors. Took on gasoline.

July 17th.—Got under way at 7:00 A. M., and arrived at Quincy at 11:00 A. M. Average revolutions, starboard engine, 490; port, 400.

COMMENTS.

From the public point of view this cruise was very satisfactory in demonstrating the ability of a vessel of this class to make an extended cruise in rough weather.

From the viewpoint of a submarine officer the cruise merely emphasizes a few facts that had been previously experienced in other submarines of similar type.

The *Salmon* in cruising rig has a positive buoyancy of 49 tons, and when heavy seas were encountered on July 8th, the conning-tower hatch was closed, and so far as water tightness is concerned the boat was sealed up with the exception of two telescoping ventilators, 6 inches in diameter, that furnished air to the engines. In case of danger from large quantities of water coming down, a valve is fitted in the base of each ventilator which closes automatically by the inrush of water. These valves are also operated by levers in case they fail to work automatically.

The vessel is steered from inside the conning tower, a lookout being kept through observation ports, and also through the forward periscope.

The conning tower is not large enough for two men to oc-

cupy it at the same time, as was shown on July 8th. The objections to a larger conning tower are that it lessens the submerged speed and that it is more difficult to give the required structural strength for deep submergence, both of which are essential features in a submarine. These difficulties should, however, be overcome in order to increase the military efficiency of vessels of this type.

The capacities of the lubricating-oil tanks and the fresh-water tanks are not commensurate with the supply of gasoline fuel, and permanent provision should be made for a greater supply of both, in order to increase the radius of action.

A submarine of 288 tons displacement, with 49 tons positive buoyancy, in light condition, is habitable and safe in rough weather, but very uncomfortable. In a gale a submarine is undoubtedly safer than any other type of vessel, due to the greater structural strength and the ability to seal the boat at will.

The engine troubles experienced by the builders on this cruise of the *Salmon* are similar to those that have been encountered in other vessels of similar type previously commissioned. It is hoped that much needed improvement in some of the details will follow from this experience of the builders.

TESTS OF AN EVAPORATOR.

By Lieut. S. M. Robinson, U. S. N., Member.

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1. The following tests of an evaporator were made at the Naval Academy in May, 1910, to determine the cost of evaporation and the advantages of operating under different circumstances.

2. The evaporator is of the Navy standard, small size, oval-section type, rated at 1,000 gallons per 24 hours.

The arrangement of piping, valves, pumps, distiller and heater is shown in the accompanying cut. It is possible to supply the evaporator with sea water, (1) from the sea suction, (2) from the discharge of the distiller circulating water direct to the evaporator, and (3) from the discharge of distiller circulating water to the evaporator through the heater placed in the vapor line between the evaporator and the distiller. The fresh-water tank is airtight, and is fitted with a safety valve, a relief valve, a vacuum gauge, and a $\frac{3}{8}$-inch pipe to the auxiliary condenser; there is a valve in this pipe.

The vapor from the evaporator can be taken direct to the distiller or through a small heater to the distiller.

All pumps are of Davidson type, and exhaust into the auxiliary exhaust line to auxiliary condenser.

The auxiliary condenser is fitted with a Davidson horizontal combined air and circulating pump.

The condensed steam from the coils was drawn off through a small Paul or D. G. C. valve, which probably allowed a small amount of live steam to pass.

3. Pressure gauges were fitted to the steam line and coils and on the shell. Thermometer wells and thermometers were

fitted in the pipe to the coil, in the sea suction, in the distiller discharge, and in the feed line near the evaporator. Thermometers were used in the barrels for measuring the distilled water and the drain from coils. All scales, pressure gauges and thermometers were tested.

4. The circulating water from the distiller was discharged through a 2-inch pipe, through a Worthington meter and a weir with hook gauge. The pressure necessary to force the circulating water through this pipe was sufficient to force the water into the evaporator so the services of the small feed pump could be dispensed with except when feeding from the sea suction. On account of this discharge pipe being so small, the circulating pumps used a large amount of steam.

The drain from coils was discharged into barrels containing cold water, and thus the weight of steam used and the temperature at which it was drained off were obtained.

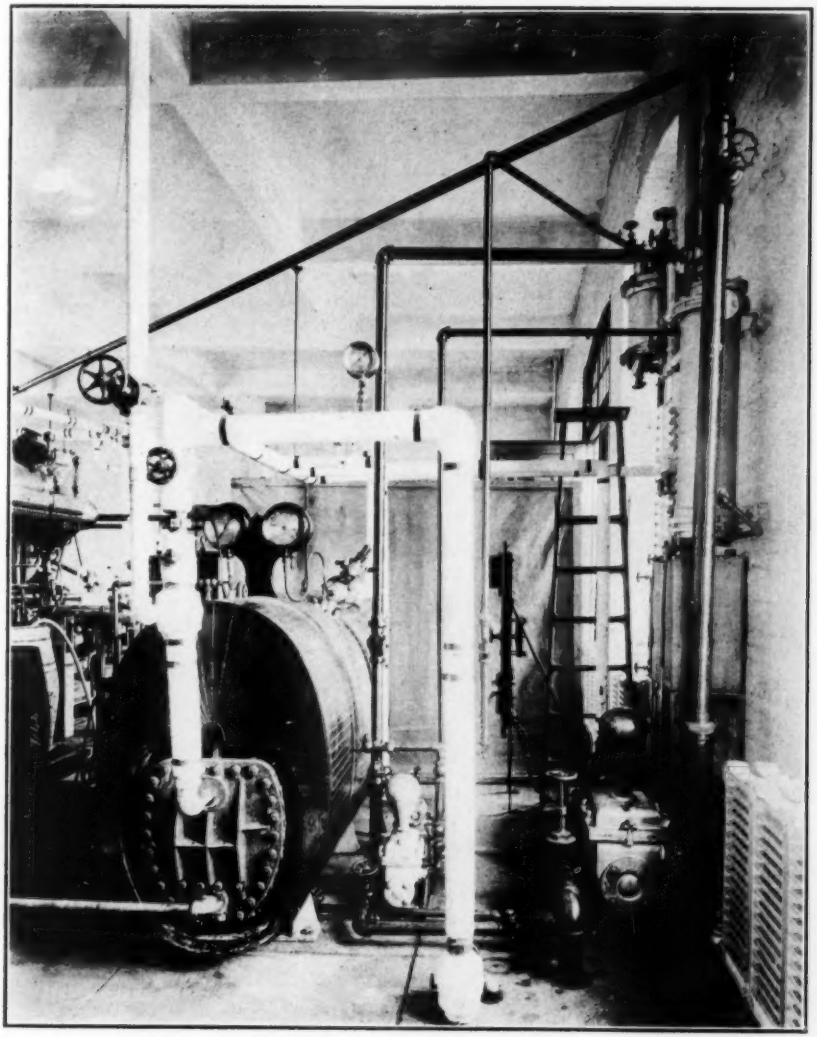
The distilled water was drawn off in a barrel, weighed, and the temperature taken.

The steam used by the pumps was obtained by weighing the water discharged from the auxiliary condenser, with only the pump in use exhausting into the condenser.

4. On account of the small size of the vapor pipe from the evaporator, it was impossible to carry a low shell pressure without at the same time carrying a low coil pressure. When a vacuum was used on the fresh-water tank it was necessary to reduce the shell pressure to prevent priming. This small vapor pipe prevented running the evaporator, at any time, up to full capacity, but the tests indicate the saving that can be made by heating the feed water and by using a vacuum on the fresh-water tank. Although none of the tests were run at full capacity, all were run at more than rated capacity.

Many preparatory tests were run in order to put all gear in good working condition and to obtain uniform conditions of running. Several of the tests were duplicated in order to insure accuracy of the data.

In all tests the start was made with the water level at a mark and everything settled down to working conditions. Data



OVAL TYPE, 100-GALLON EVAPORATOR.

was taken every half hour and the water level, and conditions at the finish were the same as at the start.

5. Test No. 1 was run with the feed water taken direct from the sea suction, and the vapor discharged direct to the distiller.

Test No. 2 was run with the feed water taken from the discharge of the distiller circulating water and the vapor discharged direct to the distiller. This test shows a decrease of 0.9 of 1 per cent, in the cost of steam for evaporation as well as a slight increase in the capacity of the plant.

Test No. 3 was run with the feed water taken from the discharge of the distiller circulating water and delivered through the heater to the evaporator, and the vapor discharged through the heater to the distiller. This test shows the great decrease of 15 per cent. in the cost of evaporation. This is quite practicable aboard ship and would greatly increase the economy, as the feed water can be heated almost to the temperature of the water in the shell of the evaporator. The feed-water heater can be made very small, and the heater, with the necessary piping and fittings, will add little to the weight and complexity of the plant.

Test No. 4 was run under the same conditions as Test No. 3, with the addition of a vacuum of $4\frac{1}{2}$ inches on the fresh-water tank. This test shows a decrease of 16.2 per cent. in the cost of evaporation, and a considerable increase in the capacity. It is doubtful whether the use of a vacuum would be of value on board ship at sea on account of the great care necessary to prevent priming and on account of the fact that the auxiliary exhaust usually carries a back pressure.

It might be used in port, where a feed-water heater is not in use.

Although the steam consumption of the pumps was measured, this is not considered important, as the amount of steam used would vary greatly, depending on the installation. This percentage of gain as given is based on the amount of steam used in the coils, and if the consumption of the pumps were also considered the increase would be greater.

Number of test	1	2	3	4
Date of test, May, 1910	5	6	9	11
Duration, in hours	4	4	3	3
Pressure, in pounds per square inch :				
Coil : Gauge	28	28	28	23
Absolute	43	43	43	38
Shell : Gauge	15	15	15.2	10
Absolute	30	30	30.2	25
Steam for pumps : Gauge	111	111	111	111
Absolute	126	126	126	126
Tank for dist. water : Vacuum, in ins.	0	0	0	4.6
Absolute	15	15	15	12.7
Temperatures, in degrees Fahrenheit :				
Coil : Inlet	271	271	271	264
Discharge	271	271	271	264
Shell : Inlet	62	82	221	210
Discharge	250	250	251	240
Circulating water to distiller: Inlet	62	62	62	62
Discharge	82	82	77	82
Cir'ing water to feed heater: Inlet	...	...	77	82
Discharge	...	...	221	210
Steam for pumps : Inlet	345	345	345	345
Dis'ge to feed tanks	108	108	108	108
Water, in pounds :				
Distilled	128	124	114	122
Circulating, discharged overboard	88,485	90,246	88,613	59,415
Total	88,485	92,015	90,136	60,722
Feed water used or water distilled	1,726	1,769	1,523	1,307
Steam used for coils	2,249	2,283	1,681	1,431
Steam used for pumps	398	308	322	231
Water evaporated and distilled per 24 hrs., gallons	1,283	1,264	1,450	1,245

	B.T.U.	B.T.U.	B.T.U.	B.T.U.
Test number	1	2	3	4
Steam used in the coil	2,620,085	2,659,100	1,958,365	1,663,394
discharged from coil	537,511	545,637	401,759	331,992
Feed water	51,780	88,450	287,847	232,646
Saved by taking feed water from distiller circu'ng water.	0	35,380	.....	.....
Saved by putting feed from distiller through heater	.....	.....	242,157	193,436
Distilled water	165,696	162,748	124,886	117,630
Circulating water discharged overboard	1,769,700	1,804,910	1,329,195	1,188,300
Steam supplied to the pumps	472,426	365,596	382,214	274,197
Pump steam to feed tank from condenser	30,248	23,408	24,472	17,556
Steam used by the pump	442,178	342,188	357,742	256,641
Steam at 43 pounds absolute :	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*
To evaporate 1 pound of sea water as feed to evaporator	1.303	1.291	1.104	1.092
absolute, to run the pumps	.220	.166	.202	.169
to evaporate and distil 1 pound of water	1.523	1.457	1.306	1.261

HEAT BALANCE.

Test number.	1				2				3				4			
Supplied to evaporate 1 lb. water at 32° F. and distil it.	B.T.U.		Per cent.		B.T.U.		Per cent.		B.T.U.		Per cent.		B.T.U.		Per cent.	
	1,821.7		100		1,759.8		100		1,725.8		100		1,660 5		100	
.....	Saved.	Lost.	Saved.	Lost.	Saved.	Lost.	Saved.	Lost.	Saved.	Lost.	Saved.	Lost.	Saved.	Lost.	Saved.	Lost.
Discharge from the coil.	311.4	...	17.1	...	308.5	...	17.5	...	263.8	...	15.2	...	254.0	...	15.3	...
Distilled water.....	96.0	...	5.2	...	92.0	...	5.2	...	82.0	...	4.7	...	90.0	...	5.4	...
Pump steam from condenser.	17.6	...	0.9	...	13.2	...	0.8	...	16.1	...	1.0	...	13.4	...	0.9	...
Circulating water....	...	1,025.3	...	56.3	...	1,020.3	...	58.0	...	872.7	...	50.5	...	909.2	...	54.7
Saved in feed water.	...	...	...	...	20.0	...	1.1	...	159.0	...	9.2	...	148.0	...	8.9	...
Steam used by pumps.	...	256.2	...	14.1	...	193.4	...	11.0	...	234.9	...	13.6	...	196.3	...	11.8
Radiation from evaporation.	...	78.2	...	4.3	...	86.4	...	4.9	...	52.7	...	3.1	...	41.7	...	2.5
Radiation from piping, etc.	...	32.0	...	2.0	...	26.0	...	1.5	...	44.6	...	2.6	...	7.9	...	0.5
Total saved.....	425.0	...	23.2	...	433.7	...	23.5	...	520.9	...	30.1	...	505.4	...	30.5	...
Total lost	...	1,396.7	...	76.7	...	1,326.1	...	75.4	...	1,204.9	...	69.8	...	1,255.1	...	69.5
Grand total.....	1,821.7		100		1,759.8		100		1,725.8		100		1,660.5		100	

NOTES ON THE TRIALS AND PERFORMANCES OF THE S. S. *OTAKI*, FITTED WITH A COMBINATION OF RECIPROCATING AND TURBINE MACHINERY.

By Engineer-Commander W. McK. Wisnom, R.N.

Read 4th August, 1909, before the North-East Coast Institution of Engineers and Shipbuilders.

In 1894 the Hon. C. A. Parsons took out a patent for the "combination" of a reciprocating engine with a steam turbine, the object of which was—"To increase the power obtainable by the expansion of the steam beyond the limits possible with reciprocating engines."

A paper on the subject by the Hon. C. A. Parsons and Mr. R. J. Walker was read at the Spring Meeting of the Institution of Naval Architects, in 1908, and the present remarks may be regarded as supplementing the reference then made to the *Otaki.*

The *Otaki* is the first merchant vessel fitted with this "combination system" of machinery. The vessel was built for the New Zealand Shipping Company by Messrs. Denny, of Dumbarton, and was delivered in November, 1908. She has since completed a voyage to New Zealand and back. She is virtually a sister ship to the twin-screw vessels *Orari* and *Opawa,* fitted with reciprocating engines, which were built by Messrs. Denny for the same company, and delivered in 1906. The only important differences in the vessels are:

(1) The length of the *Otaki* was increased 4 feet 6 inches to make up for the loss in cargo capacity due to three shaft tunnels instead of two.

(2) The design of stern and stern post was modified in the *Otaki* as necessary for three propellers instead of two.

The boiler installations in the three vessels are identical, and consist of five single-ended cylindrical boilers, fitted with Howden's forced-draft system. Each boiler has three furnaces. The total grate area is 305 square feet, and the total heating surface 13,500 square feet. The working pressure is 200 pounds per square inch.

The vessels being so nearly alike and the boilers similar, fairly accurate comparisons of the performances of the two types of machinery can be made.

The principal dimensions of the *Otaki* are:

Length between perpendiculars,	464 feet 6 inches.
Breadth, moulded, ...	60 feet.
Depth, moulded, ...	34 feet.

The deadweight capability is about 9,900 tons on a draught of 27 feet 6 inches.

The engines of the *Otaki* consist of two sets of ordinary triple-expansion reciprocating engines, driving wing propellers, and a low-pressure turbine driving a center propeller. The arrangement is shown in Figs. 1, 2 and 3.

In ordinary ahead working the reciprocating engines exhaust into the turbine, which can only revolve in one direction, but change valves are fitted, so that the reciprocating engines can also exhaust direct to the condenser. These change valves are operated by gear from the reversing shafts; the act of reversing the reciprocating engines closes the connections between them and the turbine, and opens direct communication from the reciprocating engines to the condensers. The turbine is thus automatically thrown out of action when going astern. The operating gear for the change valves is also arranged so that the turbine can be cut out at any time without stopping, and the ship can then be maneuvered as an ordinary twin-screw vessel. The arrangement also permits the turbine to be used in combination with one reciprocating engine only for ahead

working. The arrangement of change valves and gear is shown in Figs. 4 and 5.

Two condensers are fitted, and each set of reciprocating engines exhausts direct to its own condenser when the turbine is not in use. The turbine exhausts to both condensers. A shut-off valve is fitted in the exhaust from the turbine to each condenser, so that in the event of a defect in either condenser, or its pumps, or other connections, the corresponding shut-off valve can be closed, and both exhausts led to the other condenser only, thus enabling repairs to be made without stopping any engine. The shut-off valves are shown in Figs. 6 and 7.

The diameters of cylinders of the reciprocating engines in the *Otaki* are 24½ inches, 39 inches, and 58 inches respectively, with a stroke of 39 inches. The cylinders of the *Orari* and *Opawa* are 24½ inches, 41¼ inches, and 69 inches in diameter respectively, and the stroke 4 feet. The ratio of high-pressure to low-pressure cylinders is thus 1 to 5.6 in the *Otaki* and 1 to 7.93 in the sister ships fitted with reciprocating engines.

The turbine fitted in the *Otaki* is, generally speaking, of the ordinary Parsons low-pressure type. The diameter of rotor drum is 7 feet 6 inches, and the length of blades 4¾ inches in the first expansion to 12 1/16 inches in the last expansion.

A novel feature is that the drum is completely closed at both ends, and any leakage past the dummy is led away by an external pipe to the condenser. The object of this is to reduce the loss due to the cooling effect of the condenser on the internal surface of the drum, and to prevent corrosion.

The turbine spindles are packed with soft packing, separated into two parts by a metallic lantern ring. This ring enables the shaft to be surrounded by water between the two divisions of the packing, thus forming a water seal. The water is supplied under a slight head, and a gauge glass at each stuffing box indicates the head of water and whether air leakage is occurring. A full-sized model of the arrangement was kept

running for a month on shore before it was decided to adopt this method of packing which has been found to work very satisfactorily on service. The stuffing box is shown in Figs. 8 and 9.

The lubrication of the turbine bearings is on the adjustable sight-feed system, and the oil pump is worked off the turbine shaft. An alternative oil pump is worked from the air pump lever of one of the reciprocating engines.

The condensers were made to the design of the Contraflo Condenser Company, and the total cooling surface of the two condensers is 6,000 square feet. Figs. 10, 11 and 12 show the design of the condensers.

There is one single-acting air pump of the ordinary bucket type worked from each set of main engines, the diameter of the bucket being 26 inches, and the stroke 19½ inches.

The circulating pumps have 16-inch suctions and discharges, and the diameter of the impellers is 48 inches.

The feed pumps are of Woodeson's patent type.

A surface feed heater is fitted in connection with the exhaust from auxiliary engines. Fig. 13 shows the general arrangement of the feed heater.

The vessel was designed for a continuous sea speed of 12 knots when fully loaded, but the contract provided for obtaining a trial speed of 14 knots with 5,000 tons of deadweight on board. The great difference between sea-going and trial conditions must be remembered when comparing sea performances and trial results.

Tables I and II give some particulars of the results of the trials.

The *Orari* obtained a mean speed of 14.6 knots on the measured mile at Skelmorlie. The *Otaki,* under the same conditions, with apparently greater ease, obtained a mean speed of over 15 knots for a total water consumption per hour of 6 per cent. less than the *Orari.* The total water consumption per hour in the *Otaki,* at 14.6 knots, was 17 per cent. less than in the *Orari* at the same speed. The total water

consumption referred to here, and in subsequent comparisons, is that for the main and auxiliary engines.

In the *Otaki* the total water used in all the engines was measured by tanks during the trials, and the number of strokes of the feed pumps was also observed.

Table I.—Particulars of Progressive Runs on the Measured Mile at Skelmorlie, 31st October, 1908.

	Mean of 2 runs.	Mean of 2 runs.	Mean of 2 runs.	Mean of 2 runs.
Total H.P. (I.H.P. of reciprocating engines plus S.H.P. of center shaft)	6,857	5,348	4,704	3,282
Speed, in knots	15.02	14.278	13.829	12.518
Revolutions, port	103	96.2	93.1	84.6
starboard	103.5	97.9	93.5	83.4
center	224.5	209.7	197.2	172.1
Total water consumption per hour by tanks (taken from curve), pounds	82,000	67,300	60,200	44,600
Total water consumption per hour from strokes of feed pumps (taking from curve), pounds	84,700	69,300	62,300	48,200
Water per H.P. per hour by tanks, pounds	11.95	12.6	12.8	13.6
pumps, pounds	12.35	12.95	13.25	14.7
Mean absolute pressure in H.P. receivers by indicator diagrams, pounds	193	178	166	135
Absolute pressure at turbine inlet (measured by column of mercury) pounds	9.5	7.62	6.76	5.0
Vacuum at exhaust end of turbine (measured by column of mercury), inches	28.1	28.2	28.4	28.5
Vacuum in condenser by gauge, inches	28.2	28.4	28.3	28.5
Barometer, inches	36.06	30.06	30.06	30.06
Temperature of sea water, °F	56	56	56	56
circulating discharge, °F.	70	67	70	70
feed water at hot well, °F	72	70	73	74
Draught of ship, forward, feet and inches	...	18–10	...	...
aft, feet and inches	...	21–4	...	...
mean, feet and inches	...	20–1	..	...
Displacement, tons	...	11,716	...	...
Water consumption per I.H.P. per hour based on I.H.P. of *Orari*:				
Water by tanks, pounds	13.66	13.7	13.8	13.07
pumps, pounds	14.12	14.1	14.3	15.2

The water consumption as calculated from strokes of the pumps is in all cases greater than that obtained by tank measurements, the difference being about 3 per cent. at the higher speeds.

The water consumption was not measured by tanks in the

Table II.—PARTICULARS OF VARIOUS TRIALS.

	1	2	3	4	5	6
Total H.P.	6,857	7,054	6,030	5,420	4,100	3,460
Speed, in knots	15.02	15.09	14.71	14.25	13.35	11.7
Revolutions, port	103	104.9	99.7	96.7	90.1	82.9
starboard	103.5	104.5	101.1	97.5	89.6	83.5
center	224.5	228	214	203.1	179.0	169.4
Total water consumption per hour by tanks (taken from curve), pounds	82,000	83,900	75,200	66,700	...	...
Ditto from strokes of pumps (taken from curve), lbs.	84,700	86,700	77,600	68,700	56,400	49,80[illegible]
Water per H.P. per hour:						
By tanks, pounds	11.95	11.9	12.47	12.3	...	...
By pumps, pounds	12.35	12.3	12.87	12.67	13.75	14.4*
Coal consumption pr. hr., lbs.	...	...	...	...	...	4,800
Coal consumption per H.P. per hour	...	...	..	...	...	1.387
Mean absolute pressure in H.P. receivers by indicator diagrams, pounds	193	196	189	195	...	175
Absolute pressure at turbine inlet measured by mercury column, pounds	9.5	9.8	8.5	7.5	5.75 (by	5.2 gauge)
Vacuum at exhaust end of turbine measured by column of mercury, inches	28.1	28.0	28.1	28.2	...	...
Vacuum in condenser, by gauge, inches	28.2	28.2	28.35	28.4	28.75	28.1
Barometer, inches	30.06	30.0	30.0	30.1	30.2	...
Temperature of sea water, °F.	56	56	56	56	56	55
Temperature of circulating discharge, degrees F.	70	74	70	70	70	65
Temperature of feed water at hot well, degrees F.	72	74	73	73	73	70
Draught, forward, ft. and in.	18–10	18–10	...	...	...	22–6
aft, ft. and in.	21–4	21–3	...	...	...	23–10
mean, ft. and in.	20–1	20–01	...	...	...	23–2
Displacement, tons	11,716	11,690	...	...	...	13,710
Water per I.H.P. per hour based on I.H.P. of *Orari*:						
By tanks, pounds	13.66	13.8	13.65	13.7	...	...
pumps, pounds	14.12	14.25	14.1	14.1	14.54	...

* These are actual results.

Column (1) full speed on mile (mean of two hours), October 31, 1908.
Column (2) full speed on mile (mean of two hours), November 2, 1908.
Column (3) mean of two runs on mile, November 2, 1908.
Column (4) continuous run for one hour, November 2, 1908.
Column (5) continuous run for two hours, November 5, 1908.
Column (6) run from Clyde to Liverpool (14 hrs.), Nov. 21, 22, 1908.

Orari, being estimated only from the number of strokes of the feed pumps. In order to make a fair comparison of the consumption in the two ships, it is necessary to consider the water as estimated from strokes of the feed pumps in both cases, or to correct the estimated consumption in the *Orari* to make it comparable with the tank measurements of the *Otaki.* In view of the greater accuracy of tank measurements, the latter method of comparison has been adopted.

The corrected total water consumption of the *Orari,* at a speed of 14.6 knots, was 88,300 pounds per hour, or 16.5 pounds per I.H.P. per hour. The latter figure is about the mean of the results obtained under similar conditions in a number of vessels with machinery of similar design, and so may be regarded from a practical point of view as sufficiently reliable for purposes of comparison.

The records of water consumption are in most cases based on observations on the measured-mile runs, and it is admitted that such records are liable to error on account of the short duration of the observations. It has therefore been considered preferable to plot the results of various observations, to draw a curve of mean results, as shown in Fig. 14, and to use figures taken from the curve in making the comparison previously given and those which will be referred to later.

The actual curves from which the figures were taken were, of course, on a larger scale than those herewith, and care was taken in drawing the curves not to give unduly favorable results.

The following auxiliary engines were at work during the trials:

Two main circulating engines; 1 air pump for auxiliary condenser; 1 circulating pump for auxiliary condenser; 1 main feed pump; 1 fan engine; 1 electric-light engine; 1 steering engine; 1 general service pump.

The evaporators were not in use during the trials on the Clyde, and the results do not include any allowance for steam used by evaporators, or make-up feed in any case. On the other hand, it was found that all the auxiliary exhaust steam

could not be utilized for heating the feed water, so there was a certain amount of loss through taking part of the auxiliary exhaust direct to the auxiliary condenser.

The coal consumption was not ascertained on these trials. the time being too short to admit of accurate measurement.

On the run from the Clyde to Liverpool, with the vessel partly loaded, on 21st and 22d November, 1908, at about half the full power, records were taken for fourteen hours and the coal was measured.

The water was estimated from the total number of strokes of the feed pump during the period.

The evaporators were not in use.

The coal consumption works out at 1.387 pounds per H.P. per hour for all purposes, but, as will be seen from the particulars of water consumption, a lower coal consumption per H.P. might be anticipated at higher power.

The coal used was Scotch, of the following quality, as analyzed by two independent analysts:

	(1.) Per cent.	(2.) Per cent.
Carbon, ...	67.15	66.48
Gas, tar, etc., ...	24.02	24.28
Sulphur, ...	.84	
Ash, ...	6.13	7.50
Water, ...	1.86	1.74
	100	100
Calories in heat units, Cent., ...	7470	7564
Heating power in pounds of water evaporated from and at 212 deg. Fah. per pound of coal, ...	13.91	14.07

Accurate comparisons of coal and water consumption per horsepower in two vessels driven by reciprocating engines and steam turbines respectively are rendered difficult by the distinction between shaft horsepower and indicated horsepower. The actual shaft horsepower of the reciprocating engines is seldom known, and an assumed ratio of shaft horsepower to indicated horsepower, based on general information, may lead to error. Further, as the efficiencies of the propellers may differ sensibly, and we are concerned with the efficiencies of the whole propelling apparatus, the comparison of the two systems cannot be based on shaft horsepower alone.

If it be desired to compare the coal or water consumption of two vessels in relation to power, then it appears preferable to use the known indicated horsepower of the reciprocating-engined ship as a basis, provided the propellers are of normal design. We are thus eliminating any difference in efficiency of propellers, and dealing with power in the terms which custom has rendered familiar. In the case of two sister vessels the same horsepower would be used for both ships, and this method of comparison would therefore give the same results as comparison by total consumption.

From a design point of view, the most reliable comparison appears to be one based on water consumption per E.H.P. where this is available, while from a practical point of view it is obvious that in sister vessels with similar boiler installations the coal consumption per mile or per hour at the same speed affords the best comparison of the two systems.

Figure 15 shows the I.H.P. and speed curves of the *Otaki*, *Orari* and *Opawa*.

The proportion of total power developed in the turbine of the *Otaki* was found to vary with the speed. At full power this proportion was about one-third; while at very low speeds the turbine was doing only a small proportion of the work.

At a speed of 14.6 knots, the indicated horsepower in the *Orari* was 5,350, and the corresponding power in the *Otaki* was 5,880. At this speed the E.H.P. was 3,210 in the *Orari* and 3,350 in the *Otaki*, the propulsive coefficients being thus

60 per cent. and 57 per cent. in the respective ships. The propulsive coefficient in the *Otaki* at full speed fell to 54 per cent.

Applying now to the trial results of the *Otaki* and *Orari*, at a speed of 14.6 knots, the various methods of comparison referred to:

(1) The comparison based on total water consumption and speed has already been dealt with.

(2) A water consumption of 21.9 pounds per E.H.P. per hour in the *Otaki* corresponded to 27.5 pounds per E.H.P. per hour in the *Orari*, the gain on this basis being, therefore, 20 per cent.

(3) The lowest line of Tables I and II shows the water consumption per I.H.P. in the *Otaki*, taking the I.H.P. as that obtained in the *Orari* for corresponding speeds. On this basis a water consumption of 13.7 pounds per I.H.P. per hour in the *Otaki* corresponded to 16.5 pounds per I.H.P. per hour in the *Orari*, *i. e.*, the gain in the *Otaki* was 17 per cent.

A summary of the results of these comparisons is given in Table III.

Table III.—SUMMARY OF COMPARISONS OF TRIAL RESULTS OF THE "OTAKI" AND "ORARI" FOR A SPEED OF 14.6 KNOTS ON THE MEASURED MILE.

	E.H.P.	I.H.P.	Propulsive coefficient.	Water consumption.		
				Total per hour.*	Per E.H.P. per hour.	Per I.H.P. of *Orari* per hour.*
			per cent.	*pounds.*	*pounds.*	*pounds.*
Otaki.....	3,350	5,880	57	73,300	21.9	13.7
Orari.....	3,210	5,360	60	88,300	27.5	16.5
Gain per cent. in *Otaki*.....				17	20	17

NOTE.—Columns marked * do not take into account the difference of E.H.P. in the two ships; these two methods of comparison should show the same gain.

In connection with the comparisons of water consumption the results obtained in two representative vessels built for cross-channel passenger service may be of interest. One vessel was fitted with twin screws and reciprocating engines, and the other with turbines and triple screws. The reciprocating engines were of the ordinary fast-running triple-expansion type fitted in such vessels, and with a smaller ratio of expansion than usual in slower vessels. These engines were not so economical as those fitted in the *Orari*, but are believed to be economical engines of their type. The boilers of the turbine-engined ship were smaller than those of the reciprocating-engined ship, and the former obtained a slightly greater speed. The propulsive coefficient in the reciprocating-engined ship was 55.5 per cent. and in the turbine-engined ship 53 per cent. At the maximum speeds the reciprocating engines consumed 30.6 pounds of water per E.H.P. per hour, and the turbines 26.4 pounds per E.H.P. per hour. This represents a gain of 14 per cent. in consumption by the use of turbines, and it may also be incidentally remarked that the reduction in weight of machinery per E.H.P. was 18 per cent.

Table IV shows the initial pressures, range of temperature, etc., in the different cylinders as taken from two sets of indicator diagrams at full speed in the *Otaki*. It will be observed that the range of temperature in the low-pressure cylinders of the reciprocating engines is about the same as in the other cylinders; also that the theoretical heat available in the turbine is over one-third of the total available heat.

Turning now to the performance of the *Otaki* on service, the coal consumption on the voyage from Liverpool to Teneriffe was 11 per cent. less than the mean for the sister vessels *Orari* and *Opawa* under similar conditions and at practically the same speed.

On the whole round voyage from Liverpool to New Zealand and back, the average speed maintained by the *Otaki* was 11.09 knots. This speed does not represent the sea speed which the vessel is capable of maintaining, the power required being only about half that developed on trial, nor is it the

most suitable for obtaining the lowest consumption of coal per H.P., but it was the speed arranged to suit the owners' conditions of service. The initial pressure in the turbine was only about 4.5 pounds absolute.

Table IV.—INITIAL PRESSURES AND RANGE OF TEMPERATURE IN CYLINDERS OF THE "OTAKI" AT FULL SPEED.

	Reciprocating Engines.				Turbine.	
	H.P. Initial Pressure (Diagram).	M.P. Initial Pressure (Diagram).	L.P. Initial Pressure (Diagram).	L.P. Exhaust Pressure (Diagram).	Initial Pressure (Mercury Column)	Initial Pressure (Mercury Column)
Absolute pressure, pounds	196	91.5	36	11.5	9.8	1.0
Corresponding temperatures assuming saturated steam. Degrees F. (approximately)	380	321	261	200	192	102
Temperature range, degrees	59	60	61	8 (Drop)	90	
Theoretical available heat units per lb. of steam B. Th. U. (approximately)	200				120	

A careful comparison of the coal consumption for the round voyage with that of the sister ships on similar voyages at the same speed shows an apparent gain of about 8 per cent. This gain represents about 500 tons of coal for the round voyage, and as the ship leaves England with sufficient coal for the outward run, would permit of the cargo carried being increased one-half that amount. Compared, however, with the results on trial and on the voyage as far as Teneriffe, this result appears disappointing. It is, however, not really so, as there were various adverse circumstances to account for the apparent falling off in economy which it is not proposed to enter into here, as they were not attributable to the type of machinery. One contributing cause, however, was that, on docking, the vessel was found to have an abnormally foul bottom.

It is confidently anticipated that a greater gain in economy will be obtained in future.

The engines made a non-stop run outwards from Teneriffe to New Zealand, a distance of 11,669 miles as logged, which is probably the longest continuous run yet made by a marine turbine. The turbine worked perfectly satisfactorily throughout the whole round voyage.

The condensers were designed with a view to maintaining a high vacuum in tropical waters, and the following are typical records of the results obtained with sea water of various temperatures:

Height of Barometer, ins...	30.02	30.05	30.09	30.00	30.09	29.85	29.65
Vacuum, inches.....	28¼	28½	28¾	29	29	29	29
Temperature of—							
Water in hot well, °F.....	103	98	94	86	83	74	71
Sea water, °F.....	84	80	76	68	63	56	53
Circulating discharge, °F.	98	94	91	82	80	68	69
Vacuum corrected to 30-inch barometer, and reduced .35 inch to allow for possible error in gauges	27.88	28.1	28.3	28.65	28.56	28.8	29

The revolutions of the main reciprocating engines, when these records were taken, were about 81 per minute, and the revolutions of the circulating pump 150 per minute.

The vacuum is given as recorded, but a slight reduction should be made in each case for errors in gauges. The *maximum* error observed in the gauges on trial as compared with the mercurial column was .35 inch, while the mean error was under .2 inch. The gauges were not tested after arrival in England, so the *absolute* accuracy of the records cannot be vouched for.

From these records, even with a deduction corresponding to the *maximum* error observed, it appears that practically the possible vacuum with the respective hot-well temperatures was obtained. The drains from the turbine are always open to the air-pump suction, so that the temperature of the water in the hot-well was probably slightly higher than that of the water

at the bottom of the condenser. Further, there may be slight errors due to the vacuum being recorded to the nearest quarter of an inch.

The vacuum gauges are placed at about the level of the middle of the condenser.

It will be admitted, however, that even allowing a reasonable margin for errors in observation and increased error in gauges, the results appear very satisfactory, but it has to be remembered that the vessel was only steaming at about half power.

Figures for vacuum obtained on the trials are given in Tables I and II. On these trials there was air leakage at the L.P. glands which was subsequently reduced.

As regards the feed heater, with a pressure of from 8 to 12 pounds in the auxiliary exhaust, the temperature of the feed water was raised from hot-well temperature to 210–215 degrees Fahrenheit, while using only part of the auxiliary-exhaust steam. Some gain in economy would be obtained by leading the surplus auxiliary-exhaust steam to the turbine.

One of the most difficult problems in connection with the design of machinery for turbine-driven vessels is the design of propellers. It is well known that, considered from the point of view of efficiency of engines only, a high number of revolutions is most suitable for the turbine, while from the point of view of efficiency of propellers only—with the present knowledge of propeller design—a lower number of revolutions seems to be required, and turbine installations for marine propulsion are, therefore, a compromise between these two opposing factors.

In the combination system these difficulties are accentuated, as the best distribution of power has to be considered in conjunction with the effect of the difference in revolutions of center and wing shafts. It is conceivably possible that the whole gain in power obtained by the turbine may be lost by defective design or arrangement of propellers.

In this connection two recent tank experiments with models of different types of ships may be mentioned. In one case the

best results were obtained by putting a *reduced* proportion of the total power through the center screw, while in the other case the best results were obtained by putting an *increased* proportion through the center screw.

In the case of the *Otaki* a large number of tank experiments and experiments with an electrically-driven 50-foot model were made on the Gareloch before the dimensions of the propeller were decided upon.

The dimensions of the engine room in the *Otaki* were the same as in her sister ships, and were found sufficient to admit of an arrangement of machinery which gave satisfactory access to all working parts.

The total weight of machinery in the *Otaki* was about 30 tons more than in her sister ships—an increase of about 3.25 per cent. In view, however, of the greater economy of the combination system the boilers of the *Otaki* might have been reduced. Allowing for this, the total weight of an installation of the combination system would not exceed that for reciprocating engines. The saving in bunker capacity would more than balance the loss in cargo-carrying capacity due to the three tunnels, and hence in future vessels of similar design no increase in length would be necessary. In the *Otaki,* however, the owners desired no alteration in the bunker capacity or the boilers—hence the increased length.

The possible difficulty in maneuvering has been advanced as an objection to the combination system. No difficulty of any kind in connection with maneuvering was experienced in the *Otaki.* The maximum power cannot be developed when going astern, but it was found that astern power sufficient for all practical purposes had been provided.

Another objection which has been raised is the possible effect of oil from the reciprocating engines on the turbine blades, and possible clogging of the first row of fixed blades by oil and other foreign matter. Pending experience on this point it was decided not to fit oil separators or strainers in the *Otaki,* a large pocket only, with a baffle and examination and cleaning doors, being provided at the lower end of the steam-inlet pipes

to the turbine. After the trials it was found that there was no collection of oil or other foreign matter. A further examination was made at Teneriffe and another on arrival in New Zealand with similar results. On arrival in England the turbine cover and rotor were lifted and a thorough examination made. There was a slight deposit in the inlet-pipe pockets, but the blades were in good condition and quite clean, with the exception of a few in the first rows at the lowest part of the turbine on which there was a very thin layer of oily deposit. From the experience on this voyage there appears no reason to anticipate any trouble from this cause.

It is admitted that there are discrepancies in the records of the trial results of the *Otaki* and that the trials were of too short a duration for absolutely accurate measurements of water, but allowing for discrepancies and possible errors in observation, the results, on the whole, taken in conjunction with the performance of the vessel on service, are considered sufficient to show that a high degree of economy may be anticipated in the combination type of engine—an economy probably higher than has been obtained in any other type of marine engine for vessels of this class. There appears little doubt that in vessels of low speeds, for which Parsons turbines alone are unsuitable, and where economy is of primary importance, this system can be advantageously employed.

The crux of the question appears to be the design of the after body of the ship and the propellers, and the position of the propellers. With further information and experience improvements in the design of these particulars may be anticipated.

The Hon. C. A. Parsons has shown how to obtain additional power for the same steam consumption; the problem now is how to utilize this power most efficiently; in short, to make three propellers as efficient as two, and to get the full benefit of this additional power.

It is hoped, while the information given in these notes is not so complete, accurate or consistent as might be desired, that in view of the importance of the subject it is sufficient, as it stands, to be of interest.

The New Zealand Shipping Company is to be congratulated on its courage in allowing the experiment to be made, and it is desired, in conclusion, to thank the officials of the company for their courtesy in supplying much of the information contained in this paper and allowing it to be published.

APPENDIX.

Since the foregoing was written, the *Otaki* has completed another voyage to New Zealand at a mean speed of 12.35 knots, and the anticipations as regards improved economy have been realized.

As the speed was greater than that of sister vessels under similar conditions, it is difficult to make an exact comparison, but it appears that the gain in coal consumption in the *Otaki,* as compared with similar vessels, was not less than 12 to 15 per cent.

Discussion.

MR. D. B. MORISON remarked that he had come quite unprepared to take part in the discussion on the paper, but he felt that Messrs. Denny & Co., and the New Zealand Shipping Co., should be congratulated upon their courage in making this distinct advance in marine engineering, and thanked for their kindness in so unreservedly putting the results of the voyage before marine engineers. It would appear that the consumption economy available was about 12 per cent. That, he believed, had been the experience of the White Star Company with the *Laurentic,* the results being so satisfactory that they had adopted the system in their new transatlantic liners. Whether any further marked gain in economy would be achieved was doubtful, so it remained for decision whether, say, 12 per cent. was sufficient to justify the extra cost in boats of moderate speed. Personally he had been very much interested in the problem in its connection with cargo boats. Taking a cargo boat of 1,200 H.P., an exhaust steam turbine rendered 200 H.P. available, but the difficulty was to apply that 200 H.P. to the existing shaft. An additional shaft need

not be considered. Mr. Parsons had proposed gearing, belt driving and electrical driving. He himself thought that belt driving was quite out of the question. Engineers looked with a certain amount of suspicion on gearing on board a steamship, and electrical contrivances had their objectionable features. He saw in London the other day a very interesting device for magnetic driving, of the worm-and-wheel type, in which there was no mechanical contact whatever between the worm and the wheel. A difficulty presented itself if the propeller raced and the apparatus got out of phase, but no doubt further investigations would be made. In the meantime the problem was still before them of supplying, say, 200 H.P. available in a cargo boat to the main shaft, and if that problem could be solved in a simple and efficient manner, it would certainly mark a distinct step in advance in cargo-boat engineering. They were greatly indebted to Engineer-Commander Wisnom for the very detailed and expressive way in which he had put the whole results before them.

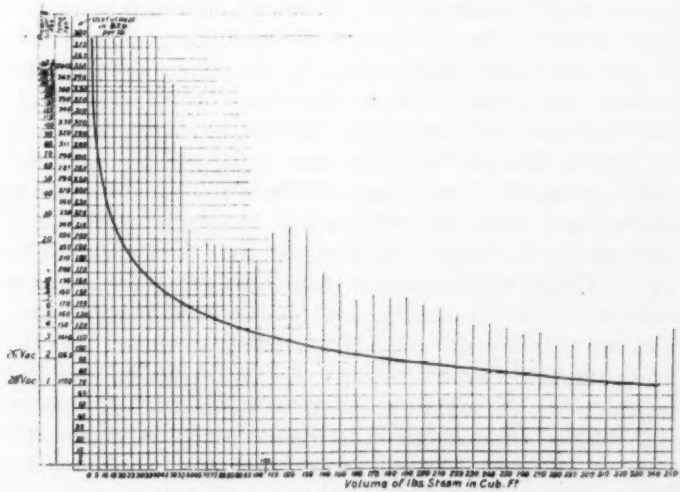


Fig. 16.

MR. W. B. SAYERS esteemed it a very great honor to make a few remarks which might be more or less critical, and especially before such an audience. First of all, he would like to refer to the curve which he had exhibited on the wall, Fig. 16. The ordinates of that curve were the thermal units available in steam from 1 pound pressure absolute, corresponding approximately to a vacuum of 28 inches up to about 190 pounds per square inch above atmosphere. The abscissae were proportional to the size of the plant that was required to obtain the modicum of power at any given point. The question that presented itself to him was, Was the combination of reciprocating engines and turbine likely to be a lasting design, or was it only transitional? He ventured to think that it was transitional. The recourse to the combination was an acknowledgment that the Parsons' form of turbine was not really satisfactory for the high-pressure end, unless for very large sizes and high speed. That was shown by considering that at a pressure of 200 pounds per square inch, and taking a steam speed of 600 feet per second, the whole of the steam used, namely 80,000 pounds per hour, would pass through a space around a drum of 7 feet 6 inches diameter of only .037 of an inch radial depth. Halving the diameter, the space through which the whole of the steam would pass would still be under one-tenth of an inch radial depth. One saw at once the difficulty that arose, there being no room for stator and rotor blades in such a space. The Parsons' form, though almost ideal at the low-pressure end, was quite unsuitable at the high-pressure end. In other forms of turbine such as the Curtis and the Rateau, the plant inefficiency or weight and bulk per unit power must be unnecessarily great owing to large sections of the rotor and corresponding portions of casing being unproductive of power. The form which promised better results was that in which the steam at the high-pressure end was led through the same rotor buckets a considerable number of times. All the stages, in fact, down to atmospheric pressure could be taken up on one rotor and ring of buckets. In such a turbine the efficiency, or steam consump-

tion attainable, was largely determined by the extent to which leakage between the rotor and stator could be minimized, and that again depended very largely upon the degree of truth to which the surface of the rotor could be made to run. Given a rotor presenting a smooth cylindrical surface, with sunk buckets running truly, the clearance could be microscopical, and the stator being in sections could be made readily adjustable while running, thus bringing the working efficiency largely under the control of the operator. He had devised a form of bucket for such a rotor which could be readily milled out of the solid at one cut per bucket, and leaving ample strength so that there was no risk of damage even should contact occur. He had made rough calculations and estimated that he could design high-pressure turbines to take the place of the reciprocating engines in the *Otaki,* to weigh not exceeding 25 tons each, including special gear to reduce the speed from 600 to 125 or 100 revolutions per minute, and thus render them suitable for equally proficient propellers to those used with the reciprocating engines. He thought the time had come when the question of gear should be practically attacked. The problem of making reciprocating engines capable of revolving at high speed with balanced cranks and rods, and gear of all kinds, having been so successfully solved, it was not too much to assume that the problem of a silent gear of ample strength and thorough reliability was practicable. He was prepared to design such a gear. He felt greatly helped by the information which the paper contained, and he was sure that Engineer-Commander Wisnom would not object to his proposal to go one notch higher.

Mr. H. A. Mavor remarked that he had not intended to speak, because one was always unwilling to side-track the discussion of a paper, but he thought that Engineer-Commander Wisnom would forgive him if he referred to Mr. Morison's suggestion of the possibility of dealing with this problem by means of electrical gear. The fact that Engineer-Commander Wisnom and others had brought skill and experience to carry

their methods to perfection, made it difficult for any electrical engineer to supersede those methods; but the kind of problem that Mr. Morison had put was one which the electrical engineer could solve with ease and certainty. To take 200 H.P. from a steam turbine and put it to drive a propeller was a problem about which there were no new difficulties. One could see all round Glasgow motors of the size spoken of running at the speed required, and controlled by the type of man who was to be found as an engineer on a cargo ship, but to add to the weights of reciprocating engine and turbine, as at present arranged, the additional weights of dynamo and motor, would be a terrible handicap, and personally he thought that this was not the way to deal with the problem. In his opinion it would be better to deal with the whole problem electrically, and this was possible on ordinarily accepted commercial lines. It was easy to show in any individual case what the probable economy would be, and he estimated that on a ship, such as the one under discussion, an economy of 10 per cent. on the coal consumption claimed by Engineer-Commander Wisnom could be attained. He begged leave to say that a visit had been arranged to the works of his company, so that anyone interested in electric ship propulsion could see in operation the methods proposed for dealing with this problem.

Mr. William Weir said that, like Mr. Morison, he considered that the marine engineering profession was once again indebted to Messrs. Denny & Co., of Dumbarton, for the first definite results of the performance of the latest and apparently most promising innovation in marine propulsion, and he looked forward to this paper as giving the lead to many shipowners, in the adoption of the combination principle. The value and interest of the paper did not appear to him to be confined to the performance of the main propelling machinery, as it contained the first public declaration of results which he had seen of the performance of a type of condensing plant which had attracted considerable attention recently, and it appeared to him that these results deserved more than passing consider-

ation in view of the great interest being taken at present in the development of the main auxiliary required for a steam turbine plant. In analyzing the results he would ask them to believe that it was his desire to establish the degree of progress in design which such figures represented. He would refer to the figures for tropical conditions later, but in the first case he was compelled to accept the trial-trip figures as being the only ones of real scientific value, as these comprised vacua recorded by mercurial columns, and accordingly might be accepted as substantially correct. Further, the trial-trip results represented four days' running over a period of almost a month, consequently allowing ample occasion for making good any considerable air leakage which developed. To render the analysis simple, he would assume that the most satisfactory measure of condenser efficiency was the number of B.Th.U.'s of heat transmitted per square foot of cooling surface per degree Fahrenheit of difference in temperature, coupled with the thermal loss in the hot-well, that was to say, the difference between the vacuum temperature and the actual hot-well temperature. In Table I, column 2, data were given which enabled one to establish a transmission rate of about 450 B.Th.U.'s, the thermal loss being 24 degrees, and the condensation rate 14 pounds of steam per square foot per hour. At the lower powers on the voyage to Liverpool the transmission rate was 200 B.Th.U.'s, thermal loss 30 degrees, and a condensation rate of 7.63 pounds per square foot. Without desiring to make any depreciatory criticism, he thought he was within the mark in saying that these figures did not represent any advance on results frequently achieved with ordinary cylindrical condensers, making no pretensions to special design: in fact, a rough comparison would show that the figures did not nearly reach results which had been already achieved and given elsewhere, but which he would briefly recapitulate as of interest to the Institution, in showing definite progress in high vacua condenser design. The first results were from a turbine-propelled vessel of about 4,500 H.P., and on careful trial the condensing plant gave a transmission rate of 960

B.Th.U.'s, a thermal loss of 20 degrees, and a condensation rate of 28.5 pounds per square foot. To this might be added the performance achieved at Clydebank shipyard, on Monday of that week, when a carefully designed condensing plant gave the following figures: Transmission rate 1,140 B.Th.U.'s, thermal loss 3 degrees, and condensation rate 18.5 pounds per square foot. In that plant the inlet temperature was 59 degrees, and the vacuum, with the barometer reading 30 inches, was 28.9 inches. He desired to state that in both those plants the proportions of the air and circulating pumps were such as could be easily fitted to any marine installation, the air-pump capacity in the first plant being one-fifth of that fitted to the *Otaki,* and in the other case less than one-half of the *Otaki.* From the shipowners' point of view it was interesting to note that the adoption of such apparatus for a vessel of the *Otaki* type, to achieve like results to those recorded on the full-power trial trip, would comprise condensers having 3,000 square feet of cooling surface, or one-half the surface at present fitted, with corresponding reduction in weight and space, while the thermal loss would be about one-third, the circulating pumps remaining of the same dimensions, and the air pumps being considerably reduced. Referring to the figures given of the performance of the *Otaki's* condensers in tropical waters, he regretted to find therein inconsistencies which rendered acceptance of the figures open to some question. The main point was that in each case the vacua, even when subject to Engineer-Commander Wisnom's deductions, were in exact agreement with the hot-well temperatures. If they accepted those figures they must inevitably accept the resulting conditions, as follows: (1) No air pump whatever was required on the *Otaki.* An ordinary air pump taking air, vapor and water could not pump any air if the air and water were at the vacuum temperature. (2) Granting there was absolutely no air leakage in the system, a most difficult presumption, a further conclusion then must be accepted, that there was no air in the steam. In view of the fact that no apparatus of the nature of a direct-contact heater was fitted

for eliminating air, this conclusion appeared quite inadmissible. (3) The condenser design shown in the paper embraced a cooler situated in the base of the condenser, and this was apparently of a non-controllable type, which could not be shut off. It therefore followed that this cooler, which was designed to perform a most important function, must have been out of action, as no cooling whatever took place. This was further proved by the evident inability of the apparatus to maintain either a steady vacuum or alternatively a steady hot-well temperature with a varying sea temperature. These functions constituted the usual claim for the adoption of the device. It must, of course, be remembered that in tropical waters the condensers were working at half the designed load. There was another peculiarity about the figures, apart from the gauge corrections, which, strangely enough, were just sufficient to make the figures theoretically possible, any lower deductions giving impossible figures. The peculiarity he referred to was that the ship had two condensers, but only one set of vacua and temperature figures were given. He would ask Engineer-Commander Wisnom to let him assume that both condensers registered the same vacua, the same outlet circulating temperature, and the same hot-well temperature. In conclusion, there was one other slight point to which he desired to draw attention. On page 54 the author said that the air pumps were of the ordinary bucket type. His recollection was that generally Messrs. Denny, in their turbine installations, had fitted suction valveless air pumps. Was he, therefore, to take it that Engineer-Commander Wisnom's experience corroborated the experience of his (Mr. Weir's) firm, in their conclusions, after careful and extensive experiments, that for efficiency, especially in high vacua plants, the three-valve type of pump was distinctly preferable? There were a number of other points in the paper which might be profitably dealt with, but he need only now assure Engineer-Commander Wisnom of his appreciation of his paper and of the results which he had placed before the meeting.

PROFESSOR R. L. WEIGHTON, M.A., D.Sc., observed that he felt personally indebted to Engineer-Commander Wisnom for this paper, chiefly because it gave what most of them interested in the economical propulsion of steamships had been in search of for the last year or two, namely, some absolutely reliable data from actual practice. He had not been able to get such data before, and this paper now gave that information on which one could rely with a very considerable degree of confidence. One point which appealed to him almost more than any other, and which was brought out in the paper, was that, it was there shown that the propulsive efficiency of the ship when propelled by the new combination of machinery fell off from 2½ to 3 per cent. He thought it was very wonderful that it was so low. In the absence of definite information, some of those interested in the combined system had been inclined to put the falling off in propulsive efficiency much higher than those figures. Fig. 15 showed that the *Otaki*, at any given speed, was consuming more horsepower than the sister ship, and, therefore, if the economy of the system were interpreted in terms of horsepower only, she would be inferior to her sister ship fitted with reciprocating engines. That inferiority, however—he took it from the few words he had had with Engineer-Commander Wisnom—arose entirely from the falling off in the efficiency of propulsion, namely, 2½ to 3 per cent., and Fig. 14 completely explained how it was that, notwithstanding that disadvantage which was inseparably associated with such machinery, the *Otaki* was nevertheless better than her sister ship. That raised the whole question of the difficulty of adequately expressing economical performance in terms of any horsepower at all. They must have it in terms of the coal consumed, or the steam consumed that gave the speed, and so it was found that in spite of the waste of power shown in Fig. 15, the previous diagram indicated superiority, because, although the combination engines wasted more power in respect of the screw propeller than the other engines did, the former gave more horsepower from each pound of steam, and that was the whole question. He

thought that point had been very clearly brought out, and if Engineer-Commander Wisnom had demonstrated nothing else than that, he had rendered a very great service indeed in this matter. Anyone in reading this paper would see at once the point of it, and he certainly would not find that the condensing apparatus was any point at all in it, and he (Prof. Weighton) would not in the ordinary course have thought it worth while to have mentioned it. He must, however, ask Engineer-Commander Wisnom to excuse him if he diverted a little very briefly from the main point of the paper, because of the remarks that Mr. Weir had made with regard to the condensing apparatus. Personally he did not know anything about the considerations which led to the adoption of this particular condensing apparatus, except from what was given in the paper, but some of Mr. Weir's remarks might be misleading unless one looked closely into the matter, and he felt compelled, in the interests of scientific accuracy and validity of comparison, to advert somewhat briefly to the question. In this connection there were only two points that he would notice. If his memory served him right, Mr. Weir compared two condensers, on a basis which was absolutely misleading as he put it, and he (Prof. Weighton) wanted to put that right in the interests of scientific accuracy. He compared two condensers which differed entirely in the conditions under which they were working. His basis of comparison was heat units transmitted per square foot per unit time, for each degree difference of temperature. Now, anyone who had experimented on the condensation of steam knew that, unless the figures obtained from such a comparison were taken in association with the conditions under which the condenser was working, the comparison was absolutely valueless as well as positively misleading. His experience was that these figures depended for their value on several factors, almost all of which were independent of the design of the condenser in any shape or form. The particular value of the figures depended first of all on the rate of condensation in the condenser. With a highly loaded condenser in which water was supplied in proportion to the steam, one would get a higher rate of trans-

mission of heat per unit time per unit surface, also if one put more water through the condenser, by increasing the speed of the water, one would equally get a higher apparent rate of heat transmission rated in the way that Mr. Weir had adopted. And further, one would get an enormous variation in that figure according to the amount of air which was permitted to enter or remain in the condenser. The whole subject was not susceptible of being stated in any such simple terms as suggested by Mr. Weir. The conditions under which the several apparatus were working must be carefully stated, and he could not possibly compare any two condensers even of the same design, not to speak of different designs, in any such simple way as that. One could not compare in that way the performance of a condenser worked as torpedo-boat condensers were worked, with a condenser worked as that in such a ship as the *Otaki.* He himself had studied this question now for several years, and he had made many hundreds of experiments both on experimental apparatus and on commercial apparatus as well, and he had found the figure given by Mr. Weir varying as follows: He had had it for *one* condenser as low as 400 and as high as 600, 563, 642, 416 and up to 1,140 B.Th.U.'s. There was no possibility of using this figure as a basis of comparison when it varied to this extent in one apparatus. He thought that if Mr. Weir's statement had been allowed to pass, some who might not have had the opportunities that others had of looking closely into the question might have been misled. The second point was that he did not know what Mr. Weir meant when he referred to the "barely possible" or "hardly possible" in the attainment of results. He would fancy that Engineer-Commander Wisnom might be left to defend the accuracy of his own figures, but if it was any use to the author, he (Prof. Weighton) might say that in his experience he had found such results as those referred to habitually obtained and maintained without any difficulty whatever, provided that the conditions were present which enabled them to be obtained. Leaving that point, one of the speakers made a remark, which struck him

as being one on which they were all agreed, that this new engine arrangement must be looked upon as transitional. In the sense that every step in advance was transitional, he agreed with him absolutely, and he thought that they all would, but if he meant that they could have no advance without finality, then he disagreed *in toto*, and so would they all. He thought that although it was transitional, and although every step in advance was transitional, it did not mean that it was not a valuable step towards final perfection. Engineer-Commander Wisnom had been kind in laying before them a distinct step in advance, though not final by any means, and possibly the lines would change on which the advance would be made in the future. For the achievement of this distinct advance they were indebted to the mechanical genius of the Hon. C. A. Parsons, and to the courage and enterprise of the owners and the builders of the *Otaki*.

Mr. D. B. Morison said that his object in asking to be allowed to speak a second time was merely to make a suggestion. He thought the greatest compliment that they could pay Engineer-Commander Wisnom and Messrs. Denny & Co. in thanking them for their generosity in publishing all the data, was to confine the discussion as near as possible to the epoch-making character of the subject. With regard to the condenser, it was simply an incident in the paper, and whether the vacuum was 29 inches or a decimal point either way made no material difference. As the subject of condensation appeared to have created considerable interest, he was certain that he was voicing the wishes of all engineers present when he asked Mr. Weir if he would favor the Scottish Institution or the North-East Coast Institution with a paper on the subject. Although Mr. Weir had a vast store of information, he had not favored—possibly through commercial pressure—the Scottish Institution, or any other Institution, lately with his views, so that such a contribution from him would be highly appreciated by the whole of the engineering profession. He (Mr. Morison) on his part would be very pleased to come

to Glasgow and join in the discussion. But if Mr. Weir found it impossible to make such a contribution he would be very pleased to do so himself. No doubt Mr. Weir would tell the Secretary which alternative would suit him, and he would leave the matter entirely in his hands.

MR. R. M. NEILSON said that in two of the columns in Table III the superiority of the *Otaki* over the *Orari* was given as 17 per cent. and as 20 per cent. respectively, and he thought that a little consideration would show that 17 was the figure of commercial importance. The *Otaki* was 17 per cent. better than the *Orari* at a certain speed, which was not the speed at which he understood the *Otaki* was intended to run, but it showed what could be accomplished by the recipro-turbine combination. A recipro-turbine-engined vessel to be of equal commercial economy would have to run on 13.7 pounds of steam per I.H.P. per hour. They, therefore, knew that with the recipro-turbine combination a vessel could be propelled at a speed of about 14 or 15 knots with 17 per cent. less coal than a fairly economical vessel propelled by reciprocating engines, and able to carry the same cargo. It did not matter commercially what the H.P. or propulsive coefficient was in the two cases, because the one vessel might have to be larger or have different lines from the other; it was the coal which was of commercial importance. Equal economy could, however, be obtained in other ways. One way would be to employ the turbo-electric drive, that was to use a high-speed steam turbine to drive a dynamo supplying current to motors which drove the screw propellers. The economy so obtained, without the use of superheated steam, would, he anticipated, be practically the same as that obtained in the *Otaki*. He referred to driving a vessel wholly—not partly—by electric transmission. Another way in which an equal, if not a better, economy could be obtained would be to retain the ordinary reciprocating direct drive and employ superheaters. In some boats engined at the Central Marine Engine Works, West Hartlepool, a considerably better economy than 13.7 pounds

of steam per I.H.P. was, he believed, obtained, although he had not the figures at hand. He did not wish to say that either of these methods of obtaining good economy was better than that employed in the *Otaki;* he only said that they deserved consideration. The recipro-turbine combination was not the only way of obtaining economy. Personally he considered that for cargo vessels the turbo-electric drive was inferior to the recipro-turbine combination, firstly, because it would be more difficult to obtain engineers to properly look after the machinery, and, secondly, because the initial cost would be much greater. Regarding the particular installation in the *Otaki,* Was the propelling machinery of the best design? Messrs. Denny deserved, he believed, great credit for the success they had achieved, seeing that they could derive no information from experience, but other engineers might have to design machinery for similar vessels and would want to know what was the best design. The *Otaki* was designed for a continuous sea speed of 12 knots. At 11.09 knots the initial pressure in the turbine was, the author said, only 4.5 pounds per square inch absolute. A high efficiency could not be expected from the turbine under those conditions. The author said that, at full power the proportion of the total work done by the turbine was about one-third, while at low speeds the turbine was doing only a small proportion of the work. Would it not have been better if the reciprocating engines had been made smaller, and more work thrown on the turbine when steaming at 12 knots? Would greater economy at 12 knots not have been obtained by so doing? The propulsive coefficient at speeds above 12 knots might have been slightly less than it was, but, on the other hand, the efficiency of the turbine, and probably also the mechanical efficiency of the reciprocating engines would have been greater than at present; and, moreover, economy would seem to be of more importance at 12 knots, the working speed of the vessel, than at higher speeds. From the particulars given in the paper it would appear that at full power the effective efficiency of the turbine, or ratio of brake horsepower to available energy

in the steam, was about 56 per cent., and at 12 knots it would probably be considerably less. The vacua attained with the contraflo condenser with high sea temperature, as given in the table, were, he considered, very satisfactory. Mr. D. B. Morison had, he knew, made a special study of obtaining high vacua with high sea temperature. The vessel was said, however, to have been steaming at only about half power when the results were obtained, so that the condenser cooling surface would be about two square feet per I.H.P. In view of the desirability of keeping down the size of the condenser, which formed a considerable proportion of the total weight of the propelling machinery, it would be interesting to know what vacuum could be obtained at full power with a sea temperature of, say, 80 degrees, or even 75 degrees Fahrenheit. At full power the cooling surface would represent about one square foot per horsepower. Or it would be interesting to know how many square feet of cooling surface would be required with condensing plant like the *Otaki's*, in order to get a 28-inch vacuum with a sea temperature of 75 or 80 degrees. It would also be interesting to know if the condensers had suffered much from oil. In a vessel propelled by turbines only, and with an auxiliary condenser, there should be no oil admitted to the main condenser. Mr. Weir had raised the question of the efficiency of the *Otaki's* condensers relatively to some other condensers. As an individual who had devoted some little time to condenser problems and who had no reason to support any particular type or make of condenser, he might say that, in his opinion, it was difficult to satisfactorily compare condensers, unless the conditions were exactly similar. He would certainly not express an opinion on the *Otaki's* condensers relatively to others, without much more data than was given in the paper. The transmission of heat through the tube sides in a condenser was influenced by many factors. One of the most important was the quantity of air present in the steam. How much air got into the condensers of the *Otaki* he did not know, but it was obvious that the air leakage conditions were very different in a vessel with a recipro-

turbine combination of machinery, to those in a vessel propelled by turbines alone.

MR. A. F. WHITE considered that if one were to judge from the mass of literature published on turbines for marine propulsion, one would think that the well-known reciprocating engine was on the eve of departure. It might be said that for the past 15 years and more the multiple-expansion engine had remained unaltered, and practically no advance beyond its size had been made in its design. Details such as pumps, packing, condensers, devices for relieving pressure on slide valves, and balancing of cranks, had effected an improvement in its working, but so far as concerned economy the engine of to-day was very little more economical than the engine of about 15 years ago. It might, therefore, be thought that so far as regarded economy the multiple-expansion engine had reached its limit. But had the multiple-expansion engine been developed to its utmost limit? It was well known that for higher pressures the turbine was not so economical as the reciprocating engine, but that for low pressures the rotary form of engine was more suited. It was, therefore, natural to suppose that a combination such as that so ably described by Engineer-Commander Wisnom was a movement in the right direction. Towards the end of his paper the author said, "Two recent tank experiments with models of different types of ships may be mentioned. In one case the best results were obtained by putting a *reduced* proportion of the total power through the center screw, while in the other case the best results were obtained by putting an *increased* proportion through the center screw." He would be pleased if Engineer-Commander Wisnom would enlarge upon that point, as it seemed to him to have a very important bearing on the question of whether more or less power should be put through the center screw. It appeared to him possible to get a higher efficiency, especially for ships steaming between 12 and 15 knots, by putting more power through the turbine, and making the center turbine to develop half the total power with high-speed

reciprocating engines on the wing shafts each developing one-fourth of the total horsepower. No doubt Engineer-Commander Wisnom had a great deal of information at his disposal, and he might see his way to supply it. They, as engineers, seemed disappointed, because they could not get their power applied in a similar manner for all classes of vessels. He thought that was a matter they should not strain after too much—for instance, taking a ship's boat of heavy construction, it would be best propelled by one man in the stern, using one oar with a screw action, but a light skiff would be propelled faster by a pair of oars at the sides, so that it might be taken as correct and natural that for ships of varying speeds different means of propulsion should be employed. Therefore, he said, perhaps the time had not yet come when the reciprocating engine could be dismissed altogether. As regarded its economy, it certainly was high time that engineers should take a step forward and make some effort to increase it, as the engine itself was eminently suitable for steamers of about 9 knots, which, after all, did the bulk of the country's trade. A good deal had been heard about the economy of condensers and devices designed to get the vacuum down to its lowest limits. That reminded him of one having a turn for economy by trying to save the candle when it was nearing the socket. A beginning should be made at the other end. The economy to be derived from the use of superheated steam was too well known to talk about to an audience of engineers. They all knew that a two-cylinder compound engine using highly superheated steam of comparatively low pressure was as economical as a triple-expansion engine with higher pressure not superheated. Engineers know that, but when the shipowner realized that an economy of at least 10 per cent., and as much as 16 per cent. could be obtained by superheating, he would insist on its use, and his engineers would then set to work and make it successful. While no engineer on shore would dream of putting down a plant without making use of superheated steam, it was strange that in the whole of the British merchant service not one engine used highly super-

heated steam, although already largely in use in foreign-owned vessels. It might, therefore, be said that it behooved engineers to no longer ignore the advantages to be derived from the use of superheated steam, and to see whether a combination of reciprocating engines and turbines using highly superheated steam could not be combined in order to produce the highest possible economy.

MR. R. J. WALKER said he would like to associate himself with the previous speakers in expressing their indebtedness to Engineer-Commander Wisnom for having placed on record in such detail the results of the performances of the *Otaki,* and to Messrs. Denny and the New Zealand Shipping Company, who had undoubtedly rendered a great service to the engineering world in allowing these results to be published. From the first introduction of the turbine in the mercantile marine, Messrs. Denny had been very closely associated with Mr. Parsons in its development, and the special facilities which that firm possessed, at Dumbarton, for obtaining scientific data, rendered Engineer-Commander Wisnom's paper all the more valuable. The statements contained therein should go a long way to remove certain doubts which existed in the minds of many engineers, as to the practical advantage to be gained in the economy in coal consumption by the combined system. In the course of his experience he had found that, whilst almost all engineers admitted that theoretically a greater efficiency could be obtained by utilizing the exhaust steam of a reciprocating engine in a low-pressure turbine, than in the case of a good quadruple engine, there were many very doubtful as to the practical utilization of this increased steam efficiency. Questions such as the reduced efficiency of the screw, of the shaft on which the turbine was placed, the loss of efficiency due to wetness of the steam entering the low-pressure turbine, the difference in thermal units expended to obtain the higher vacuum essential to the turbine end of the combination as compared with that required with reciprocating engines, and many other such points had been put forward as factors

requiring consideration in estimating the probable ultimate gain. Notwithstanding these deductions, however, there was still a very large proportion of available energy left. He thought the manner in which Engineer-Commander Wisnom had tabulated the comparative results showed conclusively the actual difference in efficiency between the sister ships referred to in the paper. He agreed with Engineer-Commander Wisnom that undoubtedly the most reliable comparison appeared to be one based on water consumption in relation to speed, as it eliminated all questions of relative efficiencies of propellers and other factors which he had already referred to. It was gratifying to note that on the basis of a given speed of vessel, the increased gain at full power was 17 per cent. in favor of the *Otaki.* Turning to the service results of the *Otaki,* it was perhaps unfortunate for the purposes of the paper that the vessel was run below her designed sea speed on her first round voyage from Liverpool to New Zealand and back. Engineer-Commander Wisnom stated that the speed of 11 knots represented about half full power, and a turbine designed for a pressure of 9 to 10 pounds absolute did not work under the most favorable conditions when running at half that pressure. He thought, however, even a gain of 8 per cent. of coal, under adverse circumstances, was well worth having, and he was confident that a greater gain would be obtained in the future when the vessel was run at her designed sea speed. Indeed, it was understood that the *Otaki* on her last voyage at a speed of 12 knots showed a further gain in coal economy. It might be of interest to refer to the results of the trials carried out by the Parsons' Marine Steam Turbine Company, in conjunction with Messrs. A. Stephen & Sons, in Sir Christopher Furness' yacht *Emerald.* In that vessel, which would no doubt be remembered as one of the first to be fitted with an all-turbine arrangement, certain modifications were made in the propelling machinery by introducing a reciprocating engine on the center shaft, in lieu of the high-pressure turbine, at the same time altering the blading arrangement of the low-pressure turbines. The ob-

ject of this modification was two-fold, firstly, to obtain a greater economy at cruising speeds, and secondly, to ascertain the possibilities of economy in coal consumption by the combination system of reciprocating and low-pressure turbines. Exhaustive trials were made towards the latter end of last year, during which the water consumption of the main engines was measured through tanks, and careful records were made of the powers developed on the respective shafts. Although there was no sister ship by which a direct comparison could be made, as in the case of the *Otaki,* the results confirmed the estimates given prior to the alteration. The shaft horsepowers developed by the low-pressure turbines at various initial pressures were as follows: With 11.6 pounds absolute pressure and a vacuum of 27.4 inches they developed 30 per cent. of the total power, the total power being the I.H.P. of the reciprocating engine plus the shaft horsepower of the turbines; with 10.6 pounds absolute pressure, and a vacuum of 27.4 inches, they developed 28 per cent. of the total power; and with 7 pounds absolute pressure and a vacuum of 27.7 inches they developed 21 per cent. of the total power. He had no doubt that the results obtained in the vessels already fitted with the combination system, and the start thus made, would lead to a more extended use of the system in the future, for the class of vessel where the designed speed fell below the limit at which an all-turbine arrangement was not considered suitable at the present time.

ENGINEER-COMMANDER WISNOM, in reply, said that he agreed with the opinion so well stated by Professor Weighton that the combination system could not be regarded as finality in marine propulsion for slow vessels. The various alternatives suggested for such vessels might show possibilities of great economy, but the point just at present seemed to be that these systems were only in the experimental stage, while the combination system had been tried practically. Regarding the condenser results, one peculiarity in the paper referred to by Mr. Weir was that the ship had two condensers, while only

one set of vacua and temperature figures was given. Mr. Weir appeared to have overlooked the fact that the ship had also two sets of reciprocating engines. In the tables the mean results only of pressure, etc., for the two sets of engines were given—it was not considered necessary for the purposes of this paper to give separate results for port and starboard engines, and the condensers were treated in the same manner. The two condensers were connected by the eduction pipes and the differences in results were unimportant. As stated in the paper, there was air leakage at the L.P. glands during the trials, and although there might have been opportunities to tighten up these glands, it was not considered advisable from the practical point of view to interfere with them. Concerning the table of condenser results on service, it was fully admitted in the paper that these were based on sea-going observations and no claim was made for their scientific accuracy. They were, however, believed to be substantially correct, and were simply inserted to show that the main object in the design of condensers—maintaining a high vacuum in tropical waters—had been obtained. No attempt was made in the design to cut down the cooling surface to the lowest possible amount. The greater power in the *Otaki,* shown by the curve in Fig. 15, in comparison with her sister vessels was not entirely due to the difference in propulsive efficiency. It would be readily seen from Table III that the E.H.P. in the *Otaki* was slightly greater than in the sister vessels. The addition in length in this case did not tend to reduce the E.H.P., and the alteration of the stern necessary for three propellers appeared to increase it. Superheating could be applied to the combination system, thus obtaining the gain of superheating at the high-pressure end, in addition to the gain by the turbine at the low-pressure end. With respect to the best proportion of power for the center propeller, that was a very wide question into which he was not prepared to enter at present. Regarding Mr. Neilson's remarks, it was hoped that the experience in the *Otaki* would lead to improved design of turbines for future

vessels. In conclusion, he desired to thank the various speakers for their complimentary remarks on the paper.

MR. WILLIAM BOYD said that it was his pleasant duty to propose a vote of thanks to Engineer-Commander Wisnom for the paper which he had read before the Joint Meeting, leading to such an interesting discussion as had taken place that day. There would be, no doubt, differences of opinion as to whether further economy could be achieved on the lines indicated in the paper that had just been read, but there could be no doubt that great thanks and great credit were due to Engineer-Commander Wisnom and those associated with him for the courage and ability with which they had put before the engineering world the results of their experience with regard to the fitting of this combination on board the *Otaki.* Whatever the lines on which the problem would be worked out in the future, there was no doubt that the information contained in the paper had conveyed a vast fund of matter for thought and consideration to all those who were interested in the great problem of economical marine propulsion.

SIR BENJAMIN C. BROWNE said that he had great pleasure in seconding the vote of thanks to Engineer-Commander Wisnom. The consideration of the economy to be derived from the combination of reciprocating engines and turbines was one of the most important questions they could have before them. Of course, they were all suffering from a depression in trade, but he hoped that before long they would all be engaged. If ever that time came it would only come in one way, and that was by working harder to grasp every source of economy, and as long as they were true to themselves in working out this problem, then he had not much fear that they would prosper in the future as they had done in the past. Reference had been made by Mr. Boyd with regard to the great kindness that the visitors had met with in Glasgow and the way in which they had been received. His first visit to the city was about 40 years ago, and since then he had always found the same

boundless hospitality. He had come in various disguises to Glasgow, in the capacity of a mechanical engineer, a naval architect, and he thought he had once come as a mining engineer, but there was always the same friendly hospitality and the same encouragement, and he always returned to the Tyne better and wiser. He had always carried back with him very happy recollections of Scottish hospitality.

The vote was accorded by general approval.

—"Trans. N. E. Coast Inst. Engineers and Shipbuilders."

Cuts showing arrangement of machinery on steamer *Otaki* will be found on pages 104 and 105 of the February, 1910, number of the JOURNAL A. S. N. E.

REPORT OF TEST OF A BABCOCK & WILCOX BOILER FOR U. S. BATTLESHIPS *WYOMING* AND *ARKANSAS*,

WITH NOTES BY

LIEUTENANT COMMANDER H. C. DINGER, U. S. N., MEMBER.

A Babcock & Wilcox boiler, representing the type proposed for installation on the battleships *Wyoming* and *Arkansas*, was tested by a Board of Naval Officers consisting of Commander C. W. Dyson and Lieutenant Commanders J. K. Robison and H. C. Dinger, U. S. N., on June 13 to 20, 1910.

The contracts for the above vessels call for economy tests of one boiler of the type proposed before installation on the vessel. The test boiler is similar in all respects to those which will be installed on the vessels, except that it is half the width. The actual boilers for the vessels will have 119 square feet of grate and 5,353 square feet heating surface, instead of 57.89 square feet of grate and 2,571.39 square feet of heating surface in the test boiler.

The contract required four tests of twenty-four hours each at successive rates of combustion of approximately fifteen, twenty-five, thirty-five, and not less than forty pounds of coal per square foot of grate surface per hour, beginning with the lowest and ending with the highest rate of combustion.

Conditions of Tests.—The tests were required to be conducted continuously, except that a maximum time of two hours will be allowed for cleaning fires after each twenty-four-hour test before beginning the next twenty-four-hour test. The only cleaning of boiler tubes allowed was by the use of steam or air lances in the same manner as is customary in actual service; the fuel to be clean bituminous coal, and allowed to be hand-picked; each test to be made at maximum

pressure of 210 pounds above the atmosphere, and the average air pressure in the fireroom at the highest rate of combustion not to exceed two inches of water.

The equivalent evaporation from and at 212 degrees F. at the highest rate of combustion of not less than forty pounds per square foot of grate surface per hour was required to be not less than eleven pounds of water into dry steam per pound of combustible.

These contract requirements were intended to represent actual service conditions as nearly as possible. The tests were arranged so that the effect due to accumulation of soot after several days' steaming would be taken into account. Thus, the results of the test at 40 pounds per square foot of grate represent the performance of the boiler when called upon for full power after several days' steaming.

The following extracts are from the report of the Board, with some additional notes. The data given is from the Board's report.

General Arrangement.—The apparatus was arranged as follows:

The boiler was erected in a sheet-iron structure, the general arrangement of which is shown in Plate I. The arrangements for supplying forced draft are also shown. The blower discharged at floor line at the back of the boiler, against a vertical baffle wall, which deflected the air upward.

The main steam pipe connected to the steam main for power house, and by a bleeder valve to an atmospheric discharge, terminating in a three-branch muffler. Both discharges were controlled by stop valves with stems extending to fireroom floor.

Feed Heater.—A Reilly feed heater was used, with a branch from the main steam pipe for supplying the necessary steam. The pressure in feed-heater shell was regulated by a valve in the steam-supply pipe. This heater was tested for tightness before and after each test.

Steam Jet.—A pipe led from the main steam pipe to a jet in the smoke pipe, with a valve for regulating the amount of opening.

Smoke Pipe.—The smoke pipe was of sheet-steel, having 19.63 square feet cross-sectional area, 100 feet in height above the grates. The location of smoke pipe with regard to uptakes is shown on sheet 1.

Calibrated thermometers were placed in the feed pipe as close to the boiler-feed stop valves as possible. Two nitrogen-filled thermometers were located in the uptakes for obtaining the temperature of the escaping gases. A thermometer for obtaining temperature of the outside air, and an aneroid barometer for indicating atmospheric pressure were hung outside of the testing room.

The draft pressures were taken through tubes inserted in the lowest dusting door of the first gas passage and in the highest dusting door of the last passage, the air pressure in compartment being measured by an air gauge hung on the bulkhead of the compartment.

Gas Analysis.—An Orsatt apparatus for making analyses of the furnace gases was located in a small compartment adjacent to the test room. The gas samples were taken from the uptake by means of a ¼-inch pipe leading to the gas-analysis apparatus. Samples of gas for analysis were taken at frequent intervals, and from the results of these analyses curves were plotted, as shown on sheets 2–7.

The tests clearly show the advantage attending the use of apparatus for gas analysis as an aid to efficient firing. When each sample was analyzed the result was marked up on a slate in the boiler compartment. Any particularly faulty condition as to air supply, clinker on grates, thick fires or holes in fires, was thus indicated and could be corrected. The curves show that the general average for percentage of CO_2 and CO is extremely good and indicate the practical possibilities of careful and scientific firing.

Method of Weighing Coal and Ashes.—A platform scale for weighing was situated in the air lock. The coal was weighed in barrows, the weight of empty barrows being carefully

checked at intervals. Each barrow was carefully balanced with a load of 200 pounds of coal. The weighing was all performed in the air lock. No inconvenience by reason of the boiler room being under pressure was experienced. At the end of each hour the floor was swept up and estimate of coal on floor made.

Method of Weighing and Pumping Water.—The arrangements for weighing water consisted of a rectangular feed tank on top of which were located two weighing tanks mounted on platform scales. The scales were tested before and during tests, and weights of empty tanks were checked at intervals. The water was led from the city main to each weighing tank, where, after being weighed, the tank was dumped into the feed tank. The feed tank was calibrated and fitted with a gauge glass so that the water could be checked at any time.

The boiler-feed pumps had a suction and an overflow pipe to the feed tank, and discharged through feed-water heater to the main feed valve on boiler. A steam gauge and water gauge, indicating steam pressure and water level in boiler, were located at the pump, thus assisting in pump regulation and in maintaining a constant feed supply. At the end of each hour the actual amount of water was checked up.

Quality of Steam—For observing the quality of the steam generated a Barrus throttling calorimeter was fitted on a branch from the main steam pipe at a point about 18 inches from the steam drum. The collecting nozzle was of standard pattern. The calorimeter was calibrated before the tests by taking readings with the pressure both rising and falling, with no steam leaving the boiler except through the calorimeter. From these standard readings the amount of moisture was calculated by the formula

$$Q = \frac{.48\,(T - t)}{L} \times 100,$$

in which Q = per cent. of moisture, T = calibration reading of the lower thermometer, t = test reading of the lower thermometer, L = latent heat of the steam at the boiler pressure.

Moisture in Coal.—Samples of coal were taken for each test, preserved in airtight jars, and from these the moisture in the coal was determined by laboratory test.

Condition of Fires and Ash Pans at Beginning and End of Tests.—The depth of fires was judged by a member of the Board at the start and at the end of each test, the fires having been brought to similar conditions at the beginning and the end of each test.

Ash pans were clean at the beginning of each test, and were cleaned at the end of the test. Water was used very sparingly in the ash pan, but was always used at the times that fires were cleaned. Fires were cleaned on each of the twenty-four-hour tests shortly before the end of test in order to enable them to be brought as nearly as possible to the same condition as at the beginning.

The Tests.—Six test runs were made, the first four being those required by the contracts for the *Wyoming* and *Arkansas*, at rates of combustion of 15, 25, 35 and 40 pounds of coal, respectively, per square foot of grate, each test being for twenty-four honrs.

The fifth test was a test for the maximum capacity of the boiler, and continued for three hours.

The sixth test was made at the rate of about 45 pounds per square foot of grate per hour, to determine the effect of lowering the back end of grate six inches below the position in the previous tests.

Coal.—The coal was hand-picked Pocahontas of excellent quality. It burned freely and clinkered very little. The coal used on all the tests was from the same lot.

Firemen employed.—The firemen employed were ordinary marine firemen picked up on the Hoboken docks for the time being, and were given no special training for the tests or for this particular type of boiler. Several firemen were discharged for drunkenness and others taken on during the tests. One water tender, two firemen and two coal passers were on duty in each shift.

The operation of the boilers and of the firing was supervised

by the Company's engineers, who stood watches in three shifts of eight hours each.

The excellent evaporative results obtained indicate the high efficiency possible with the ordinary run of firemen, if properly supervised and directed. This supervision provided for careful and regular firing, keeping fires level and not over eight inches thick, with proper use of the slice bar and leveling hoe. Special attention was paid to making all the boiler casings airtight.

Methods of Starting and Stopping Tests.—In the first test the alternate method of starting and stopping the test was employed. In the second and all succeeding tests the flying start was employed, it being much more satisfactory under the conditions required by the contract than either the standard or the alternate methods.

Gas analyses were taken by one of the Company's engineers, closely observed by members of the Board. All other data were taken simultaneously by representatives of the Company and by one member of the Board and the assistants to the Board.

Test No. 1.—At 15 pounds per square foot of grate per hour.

Begun at 6:08 P. M., June 13; finished at 6:08 P. M., June 14, 1910.

Weather during the test, warm and clear.

The blower was not run, the compartment was open, and the steam jet closed.

The evaporation from and at 212 degrees F. per pound combustible was 12.15.

Test No. 2.—At 25 pounds per square foot of grate per hour.

Begun at 8:30 P. M., June 14; finished at 8:30 P. M., June 15, 1910.

The steam jet was partly open. The forced-draft blower was used to ventilate the fireroom, but the compartment was not closed.

Weather during the test was warm and clear, slightly cloudy at the end of the test.

The evaporation from and at 212 degrees F. per pound of combustible was 12.07.

Test No. 3.—At 35 pounds per square foot of grate per hour.

Test begun at 10:25 P. M., June 15; finished at 10:25 P. M., June 16, 1910.

The steam jet was partly open. The forced-draft blower was in operation and the compartment closed.

Weather during test, cloudy and rainy.

Rate of evaporation from and at 212 degrees F. per pound of combustible was 11.77.

Test No. 4.—At 40 pounds per square foot of grate per hour.

Test begun at 11:30 P. M., June 16; finished at 11:30 P. M., June 17, 1910.

The steam jet was partly open. The forced-draft blower was in operation and the compartment closed.

Weather during the test, rainy during the first twelve hours, clear the last twelve hours.

The evaporation from and at 212 degrees F. per pound of combustible was 11.89.

Test No. 5.—At maximum capacity.

This test was conducted on June 18, 1910.

The fires were lighted with the boiler in following condition:

Temperature of water in boiler, 102 degrees; water level in glass, 1½ inches; furnaces primed.

Lighted fires at 9:04 A. M.

Steam formed at 9:12 A. M.

Steam pressure 50 pounds, 9 h., 16 m., 30 s., A. M.

Steam pressure 100 pounds, 9 h., 18 m., A. M.

Steam pressure 150 pounds, 9 h., 18 m., 55 s., A. M.

Steam pressure 200 pounds, 9 h., 19 m., 45 s., A. M.

Steam at 200 pounds in 15 m. 45 s.

The test began at 9:38 A. M., and ended at 12:38 P. M.

The steam jet was wide open. The forced-draft blower was in operation, and the compartment was closed.

At the end of this test the fires were hauled and boiler

carefully examined. There were no signs of any leaks and no distortion of any kind, either in tubes, baffles or casing, was noticed.

The weather during the test was clear.

The evaporation from and at 212 degrees F. per pound of combustible was 10.33.

Test No. 6.—At rate of 45 pounds per square foot of grate per hour, with the back end of grate lowered six inches below the level at which it was carried during the previous tests.

Test begun at 10:00 A. M., June 20; finished at 4:00 P. M., the same day.

The weather during the test was clear.

The evaporation from and at 212 degrees F. per pound of combustible was 11.30.

The tabulated data of the different test runs are shown on forms attached, and the curves of evaporation plotted from these results are shown on sheet 8.

The heat balance is shown on sheet 9.

The curve of rate of evaporation appears to show that there is a very small falling off in efficiency until a consumption of about 45 pounds of coal per square foot of grate per hour is reached, and that the boiler is almost as efficient at high rates of combustion as at moderate rates under natural draft. This is a highly gratifying result and indicates that the system of baffling is very efficient.

The difference in evaporative results, with the original grate and with the back of grate lowered, is insufficient for determining whether there is any actual advantage or disadvantage due to the lowering of the grates, when coal alone is used for fuel.

The Board reports that the sample boiler tested, representing the type of boiler to be supplied for use on board the U. S. S. *Arkansas* and the U. S. S. *Wyoming*, has fully met all requirements of the contract as to evaporative efficiency, and recommends the approval of this type of boiler for general use in the naval service.

REMARKS.

The firemen, while completely unskilled, so far as test boiler firing was concerned, soon became very much interested in the results of the gas analyses, and realized the value of so firing as to maintain as high a percentage of CO_2 as possible. This interest manifested itself very early in the tests, in the decreased density of the smoke escaping from the stack.

Particular attention is called to the excellent results obtained with this boiler under the maximum rate of combustion obtained, which slightly exceeded seventy (70) pounds of coal per square foot of grate per hour. The boiler under this condition steamed very freely, with no appreciable increase in the wetness of the steam, while the falling off in efficiency under all test conditions, from the lowest rate of combustion to the highest, was small and very regular.

After the completion of the tests the boiler was opened, cleaned and thoroughly inspected for deterioration. Not a tube showed any signs of distortion, all tubes and headers were perfectly free from blistering, and all baffles were still properly placed. This result would seem to indicate that when a boiler of this type is clean, free from scale, and built of proper material it is perfectly safe under all regular rates of combustion, up to the highest rate to which it was subjected in these tests. This finding, necessarily, supposes intelligent supervision and proper regulation of feed water during such rates of combustion.

Description and Dimensions of Boiler and Appurtenances.

Diameter of drum, inches		0-42
Length of drum, feet and inches		10-03½
Tubes, number, 560;	outside diameter, inches	0-02
	length, feet and inches	8-02
	thickness	No. 10 B.W.G.
16;	outside diameter, inches	0-04
	length, feet and inches	8-02
	thickness	No. 6 B.W.G.

Tubes, number, 16; outside diameter, inches	0–04
length, feet and inches	6–04
thickness	No. 6 B.W.G.
Furnace, kind of	Single, full width of boiler.
length, average, feet and inches	7–05
width, feet and inches	8–03½
height, average, feet and inches	3–01
Grate surface, length, feet	7–00
width, feet	8–03½
area, square feet	57.89
Heating surface, area, square feet	*2,571.39
ratio to grate	44.4 : 1
Grate bars, kind (double)	U. S. Navy Standard Bar.
width of air spaces, inch	0–00½
ratio of grate to air space	1 : .427
Smoke pipe, area, square feet	19.63
height, feet	100–00
ratio to grate	1 : 2.94
Water space, cubic inches	301,561.0
Steam space, cubic inches	102,889.0
Weight of boiler and all fittings except uptakes and smoke pipe :	
Without water, pounds	58,900.0
Water, pounds	11,000.0
Total with water, pounds	69,900.0
Total weight per square foot of grate surface, pounds	1,209.3
heating surface, pounds	25.4
Blower engines, kind	Simple, single-cylinder.
dimensions of cylinders, inches	10 × 10
fan, kind	Sirocco.
diameter, feet and inches	4–02
width, feet and inches	2–05
Area of blower inlet, square inches	1,164.15
outlet, square inches	1,590.43
Feed heater, kind	Reilly.
pumps, kind	Two Worthington Duplex
dimensions of cylinders, inches	10 × 6 × 10.

* The heating surface given is that exposed to radiant heat and to direct impact of the gases and does not include such parts of tube or other surfaces as are covered by brick work.

NUMBER OF TEST	1	2	3	4	5	6
1. Date of test	June 13-14	June 14-15	June 15-16	June 16-17	June 18	June 20
2. Duration of test....hrs.	24.06	24.5	24	24	3	6
3. Kind of fuel	Pocahontas, hand picked, excellent.					
4. Kind of start	Alternate	Flying	Flying	Flying	Flying	Flying
5. State of weather	Clear	Clear Partly Cloudy	Flying cloudy & Rainy	Flying Rainy Partly clear	Clear	Clear.
AVERAGE PRESSURES.						
6. Barometer....ins.	30.06	29.91	29.80	29.83	29.62	29.82
7. Steam pressure by gauge....lbs.	202.0	201.4	202.8	201.6	200.4	200.6
8. Force of draft at base of pipe—ins. of water, gage reading. 3rd. Pass	.31	.94	1.71	2.21	2.21	2.06
9. Force of draft in furnace—ins. of water, gage reading. 1st. Pass	.16	.46	.75	1.04	4.70	.52
10. Force of draft in ash pit—ins. of water	0	0	1.41	1.95	3.00	1.78
11. Revolutions of blower						
AVERAGE TEMPERATURES.						
12. External air....degrees F.	82.	74.	67.	69.	84.	90.
13. Fireroom....do.	93.7	97.6	88.	94.	106.	111.
14. Steam....do.	388.3	388.1	388.7	388.2	387.7	387.9
15. Feed water entering heater....do.						
16. Feed water entering economizer..do.						
17. Feed water entering boiler....do.	211.6	211.4	207.8	204.1	194.4	200.9
18. Air entering ash pit....do.	93.7	97.6	86.	94.	106.	111.
19. Escaping gases from boiler....do.	491.	545.	602.	628.	659.	604.
20. Escaping gases from economizer .do.						
FUEL.						
21. Kind of						
22. Weight of wood used in lighting fires....lbs.						
23. Weight of coal as fired*....lbs.	21200	35180	49300	57700	12200	15218
24. Moisture in coal....per cent.	.88	.74	.74	1.06	.75	.75
25. Weight of dry coal consumed....lbs.	21013	34920	48935	57088	12108	15104
26. Weight of ash and refuse....lbs.	1261	1338	1607	2477	1035	743
27. Weight of combustible consumed...lbs.	19752	33582	47328	54611	11073	14361
28. Per cent of refuse in dry coal	6.00	3.83	3.28	4.34	8.55	4.92
FUEL PER HOUR.						
29. Coal consumed per hour....lbs.	881	1436	2054	2404	4066	2536
30. Dry coal consumed per hour....lbs.	873	1425	2039	2379	4036	2517
31. Combustible consumed per hour....lbs.	821	1371	1972	2275	3691	2393
32. Coal consumed per hour per sq. ft. G. S....lbs.	15.22	24.81	35.48	41.53	70.24	43.81
33. Dry coal consumed per hour per sq. ft. G. S....lbs.	15.08	24.62	35.22	41.10	69.72	43.48
34. Combustible consumed per hour per sq. ft. G. S....lbs.	14.18	23.68	34.07	39.30	63.77	41.34

*Including equivalent of wood used in lighting fires.

NUMBER OF TEST	1	2	3	4	5	6
35. Coal per hour per sq. ft. H. S. lbs.	.343	.559	.799	.935	1.581	.986
36. Dry coal per hour per sq. ft. H. S. lbs.	.340	.554	.793	.925	1.569	.979
37. Combustibles per hour per sq. ft. H. S. lbs.	.319	.533	.767	.885	1.435	.932
QUALITY OF STEAM.						
38. Per cent of moisture in steam	0	0	.086	.172	.430	.659
39. Degrees of superheating						
40. Quality of steam (dry steam = 100)	100.	100.	99.914	99.828	99.57	99.341
WATER.						
41. Total weight of water fed to boiler* lbs.	228095	385264	527804	613754	106964	153571
42. Water actually evaporated, corrected for quality of steam (40 by 41) lbs.	228095	385264	527450	612698	106504	152559
43. Factor of evaporation	1.052	1.052	1.056	1.060	1.069	1.064
44. Equivalent water evaporated into dry steam from and at 212° (42 by 43) lbs.	239956	405298	556987	649460	113853	162323
WATER PER HOUR.						
45. Water evaporated per hour, corrected for quality of steam lbs.	9480	15725	21978	25530	35501	25426
46. Equivalent evaporation from and at 212° lbs.	9974	16543	23208	27060	37951	27054
47. Equivalent evaporation from and at 212° per sq. ft. G. S. lbs.	172	286	401	468	656	467
48. Same per sq. ft. of heating surface lbs.	3.88	6.43	9.03	10.52	14.76	10.52
ECONOMIC RESULTS.						
49. Water apparently evaporated under actual conditions per lb. of coal as fired (41÷23) lbs.	10.76	10.85	10.71	10.64	8.77	10.09
50. Apparent equivalent evaporation from and at 212° per lb. of coal (including moisture) 44÷23 lbs.	11.32	11.52	11.30	11.26	9.37	10.67
51. Equivalent evaporation from and at 212° per lb. of dry coal (44÷25) lbs.	11.42	11.61	11.38	11.38	9.45	10.75
52. Equivalent evaporation from and at 212° per lb. of combustible (44÷27) lbs.	12.15	12.07	11.77	11.88	10.33	11.30
EFFICIENCY.						
53. Efficiency of boiler; heat absorbed by the boiler per lb. of combustible divided by the heat value of one lb. of combustible. (*See* Sheet No. 6)	74.38%	73.35%	72.06%	72.80%	63.25%	69.18%
54. Efficiency of boiler, including grate; heat absorbed by the boiler per lb. of dry coal, divided by the heat value of one lb. of dry coal. (*See* Sheet No. 6)	72.50%	73.70%	72.24%	72.24%	60.00%	68.25%
REMARKS AND OBSERVATIONS.						
55. Principal data taken every		Half	Hour.			
56. Percentage of smoke as observed						
57. Method of observing same						
58. Kind of firing (spreading, alternate, or coking)	Spreading; fires level, doors fired in rotation					
59. Average thickness of fires	6-7	6-8	7-10	7-10	13-14	7-10
60. Average intervals between firings for each furnace during time fires were in normal condition	3 min.	3 min	2½ min	1½ min	1½ at start 1 at end	2½ min.
61. Average interval between times of breaking up		As required				
62. Efficiency of firemen; expert, average, or poor	Average marine firemen.					

*Corrected for inequality of water level and steam pressure at beginning and end of test.

FUEL AND GAS ANALYSES.

PROXIMATE ANALYSIS OF FUEL.

	Coal.	Combustible.
	Per cent.	Per cent.
Fixed carbon	75.26	79.00
Volatile matter	20.27	21.00
Moisture	.86	
Ash	3.61	
Total	100.00	100.00
Sulphur separately determined:	.71	.75

ULTIMATE ANALYSIS OF DRY FUEL.

	Coal.	Combustible.
	Per cent.	Per cent.
Carbon (C)	87.51	90.57
Hydrogen (H)	4.74	4.91
Oxygen (O)	2.75	2.85
Nitrogen (N)	.92	.95
Sulphur (S)	.70	.72
Ash	3.38	
	100.00	100.00
Moisture in sample of fuel as received		

ANALYSIS OF ASH AND REFUSE.

	1	2	3	4	5	6 Per cent.
Carbon Combustible	31.93	57.52	57.48	60.99	70.72	70.72
Earthy matter	68.07	42.48	42.52	39.01	29.28	29.28

CALORIFIC VALUE OF FUEL.

Kind of calorimeter used

Calorific value by calorimeter, per pound of dry coal Mean 15273 B. T. U.

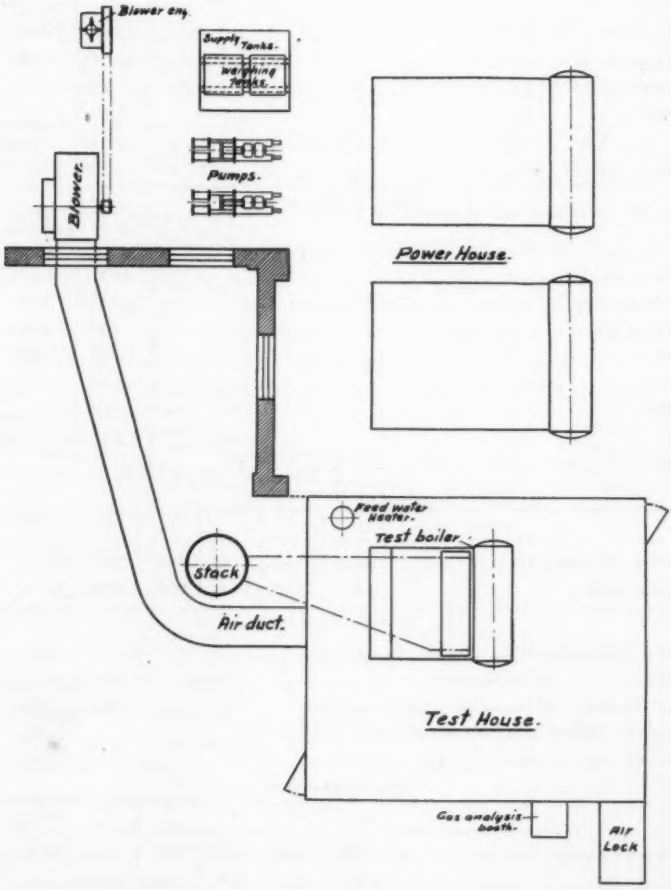
Calorific value by calorimeter, per pound of combustible 15838 Do.

Calorific value by analysis, per pound of dry coal Do.

Calorific value by analysis, per pound of combustible Do.

ANALYSES OF DRY GASES.

	1	2	3	4	5	6 Per cent.
Carbon dioxide (CO_2)	13.2	13.6	12.9	13.6	11.6	12.7
Oxygen (O)	4.1	4.7	4.5	3.9	5.07	3.0
Carbon monoxide (CO)	0.5	0.4	0.5	0.8	1.09	0.9
Hydrogen and hydrocarbons						
Nitrogen (N) (by difference)	82.1	81.3	82.1	81.7	82.24	83.4
						100.00



ARRANGEMENT OF TEST BOILER AND TEST HOUSE.

Babcock & Wilcox Co.

Plate I.

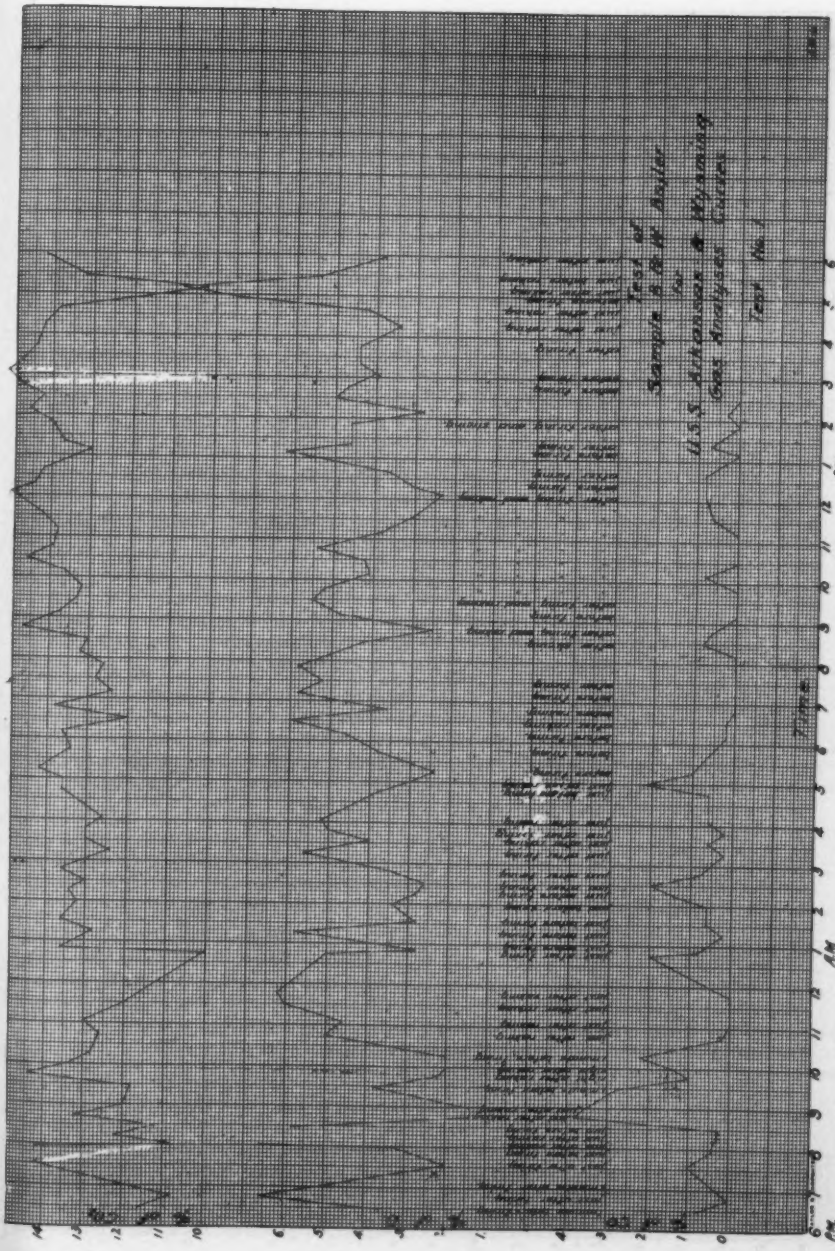
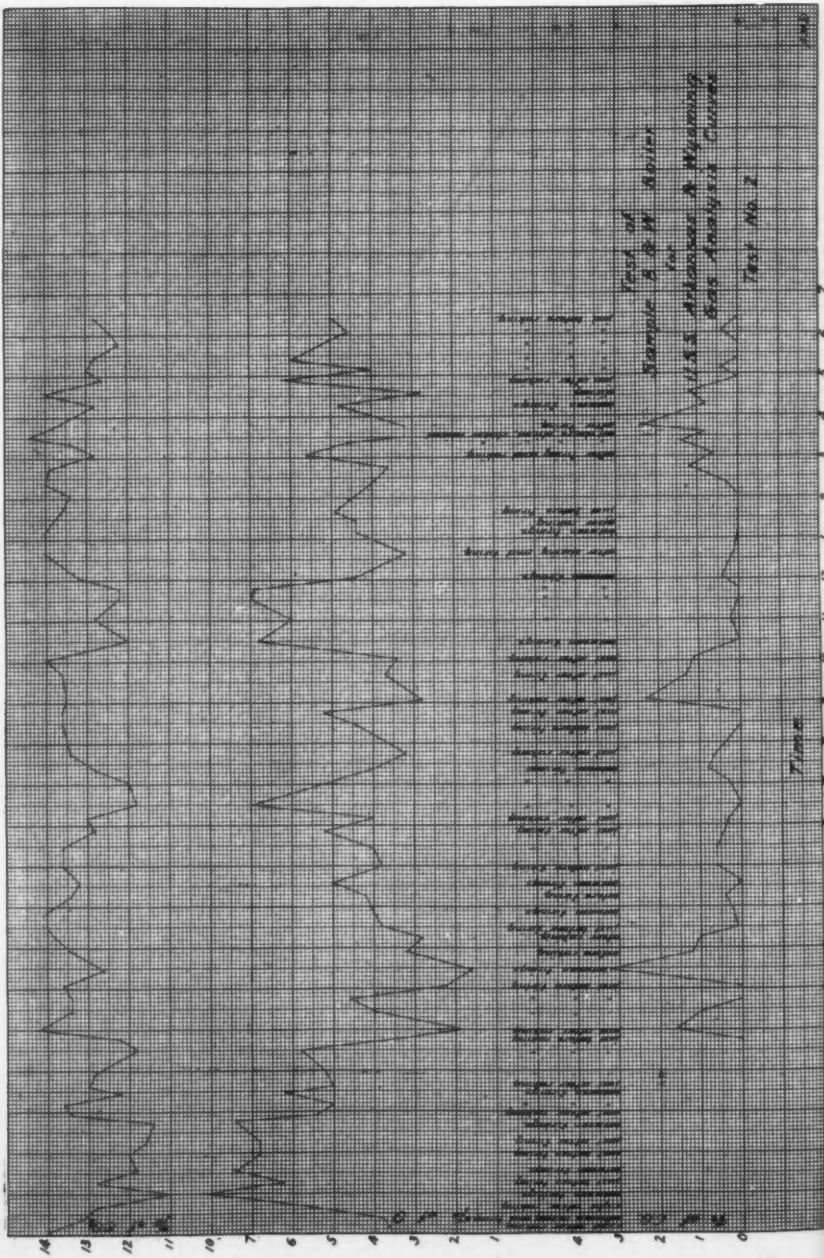


Plate II.—Test No. 1.


Test of B & W Boiler
for
U.S.S. Arkansas & Wyoming
Gas Analysis Curve
Test No. 2
Time

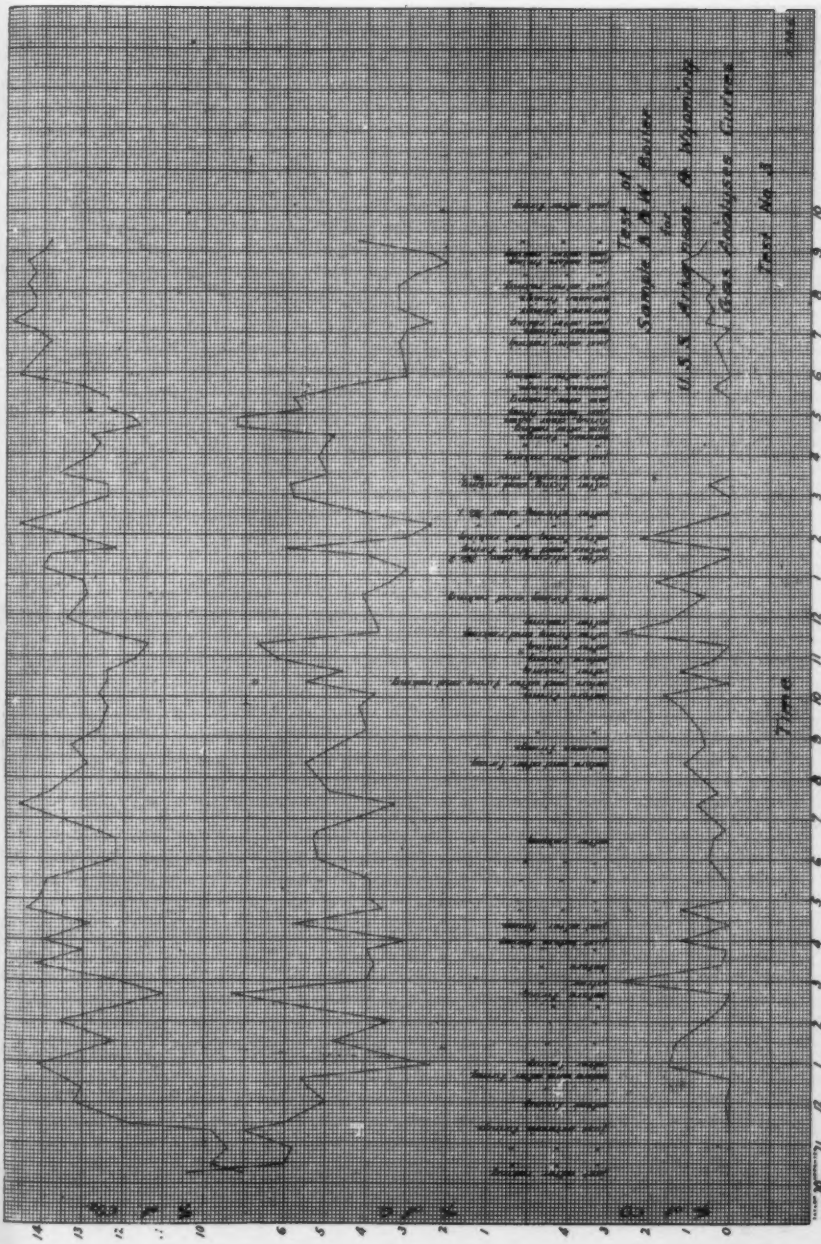
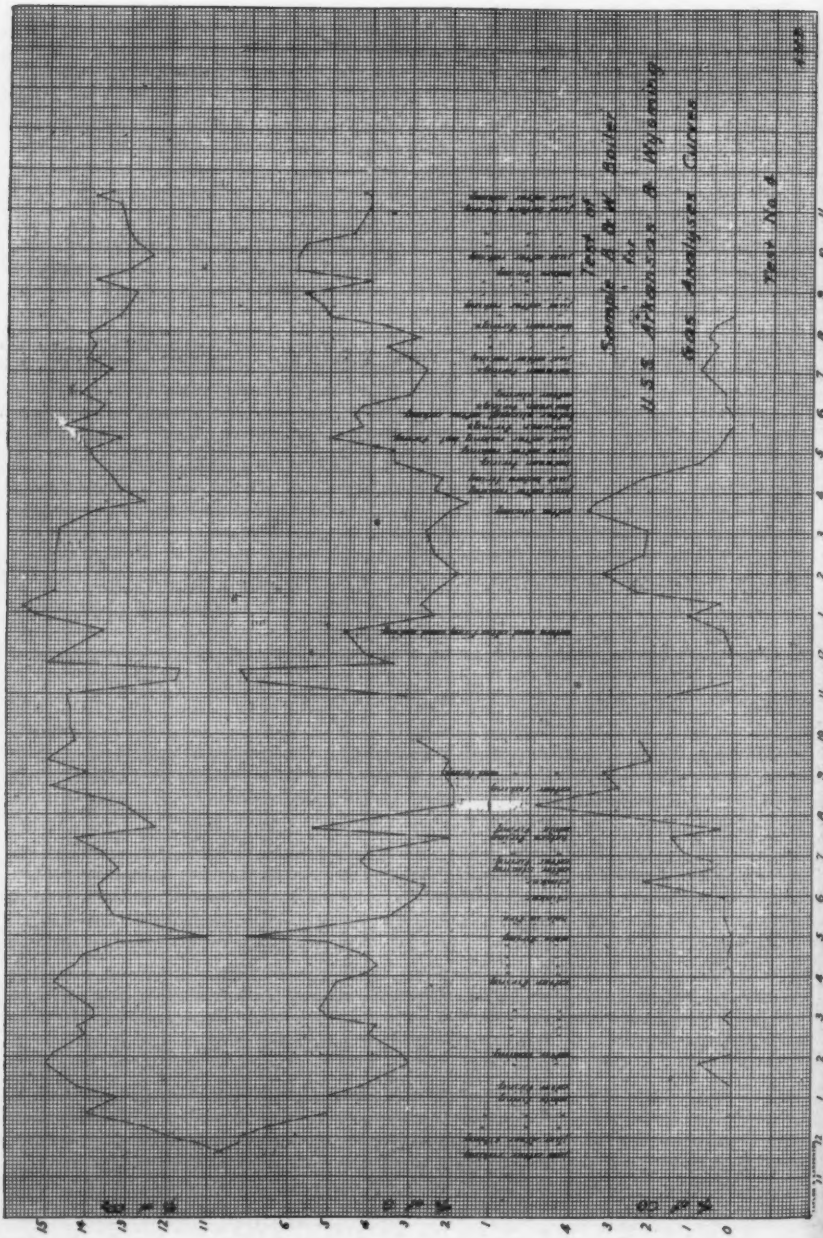


Plate IV.—Test No. 3.


Test of
Sample B & W Boiler
for
U.S.S. Arkansas & Wyoming
Gas Analysis Curves
Test No. 4

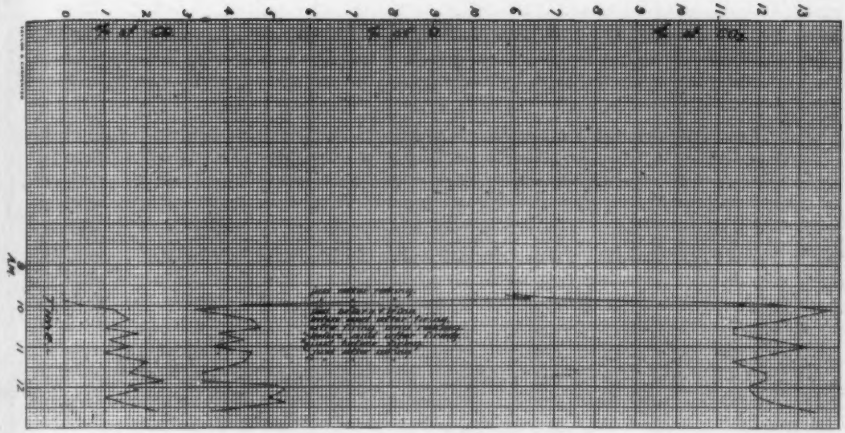


Plate VI.—TEST NO. 5.

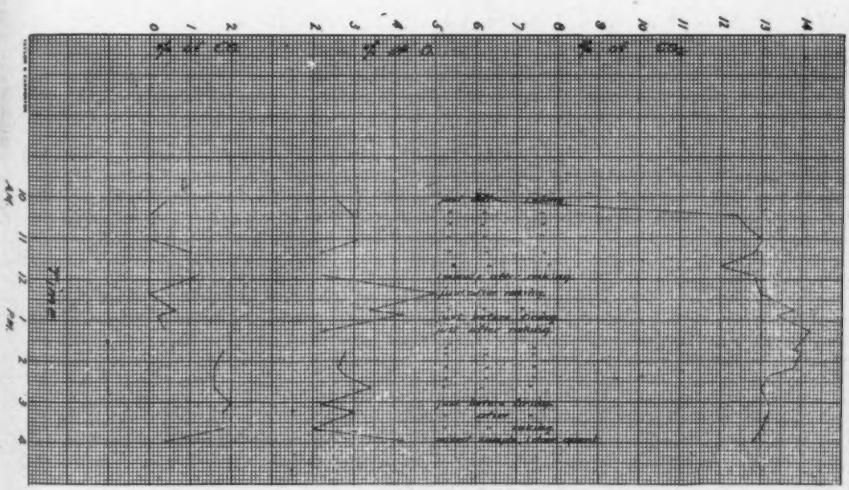


Plate VII.—TEST NO. 6.

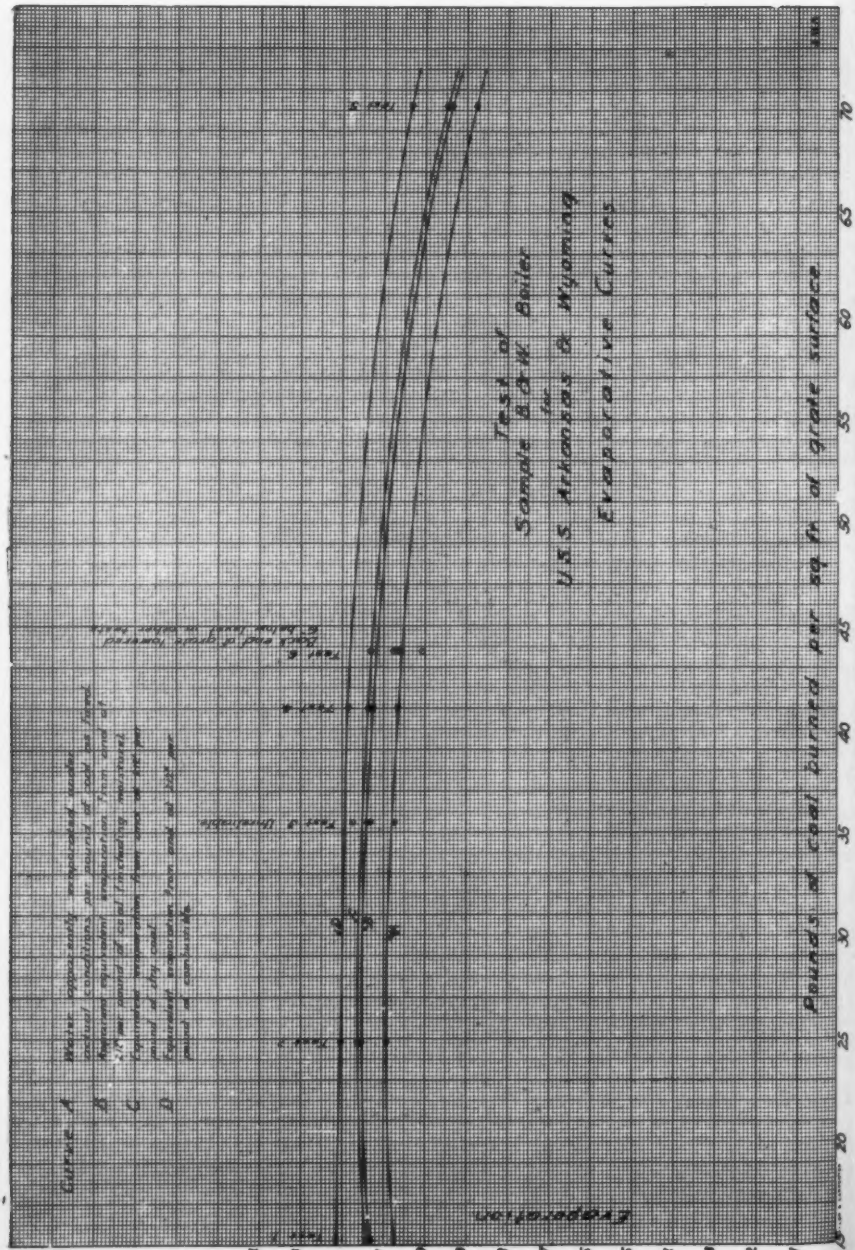

Test of Sample B & W Boiler
U.S. Arsenal Ft. Wyoming
Evaporative Curves
Pounds of coal burned per sq. ft. of grate surface
Evaporation
Plate VIII. EVAPORATION CURVES.

Plate VIII.—Evaporation Curves.

Heat Balance Sheet. B. & W. Boiler Tests June 13 to 20, 1910.

Tests of B. & W. Boilers for U.S.S. Arkansas and Wyoming.												
Number of Test.	1		2		3		4		5		6	
Pounds of dry gas per pound of carbon.	18.41		18.07		18.8		17.54		13.4		18.46	
Heat absorption and losses in B.T.U. or in per cent of calorific value of combustible.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per Cent.
Heat absorbed by the boiler per pound of combustible.	11782	74.35	11713	73.95	11413	72.06	11531	72.80	10018	63.25	10958	69.18
Loss due to moisture in coal.	10.72	.07	9.17	.06	9.45	.06	13.6	.09	9.48	.07	9.41	.07
Loss due to moisture formed by the burning of hydrogen.	519	3.27	529	3.34	545	3.44	547	3.45	549	3.46	535	3.37
Loss due to heat carried away in the dry chimney gases.	1756	11.09	1939	12.24	2317	14.63	2248	14.2	2582	16.31	2627	16.59
Loss due to the incomplete combustion of carbon.	331	2.09	257	1.61	340	2.15	505	3.18	790	4.98	606	3.83
Loss due to unconsumed hydrogen etc., to heating moisture in air, to radiation, etc.	1439.28	9.08	1390.83	8.80	1213.55	7.66	993.4	6.28	1883.52	11.93	1102.59	6.96
Calorific value per pound of combustible.	15838	100	15838	100	15838	100	15838	100	15838	100	15838	100
Efficiency of boiler per cent.	74.39		73.95		72.06		72.80		63.25		69.18	
Efficiency of boiler and furnace per cent.	72.50		73.70		72.24		72.24		60.00		68.25	

Plate IX.—Heat Balance.

THE IMPROVEMENT IN DESIGN OF CONDENSING INSTALLATIONS ON BOARD OF SEA-GOING VESSELS.

By Hans Vogel, Stettin.

Translated by Wm. Wachsmann, Associate.

(Continued from Vol. XXII, No. 1, February, 1910.)

III.—THE CIRCULATING PUMP.

On nearly all ships the circulating pump serves a double purpose; it is a circulating pump, drawing water from the sea, and it also serves as a bilge pump, drawing water from the bilge. The condenser circulating pump in a modern marine installation should be designed having the following points in view:

(1) Maximum economy of steam consumption.

(2) Greatest possible efficiency at various speeds.

(3) Compactness, so as to have the smallest possible weight and occupy the least space.

(4) Simplicity and safety in operation.

(5) Reduction in the attendance required for safe operation.

(6) Lowest cost.

(7) On account of the great sensitiveness of a condenser during the admission of air a continuous stream of water, if practicable, must be produced to prevent the tubes becoming leaky through shockwise admission of water.

From these considerations we may reasonably enter upon the question, How must the best all-around pump be designed?

(1) To obtain the smallest possible dimensions a high velocity of flow must be chosen. This velocity, however, is again dependent on the peripheral velocity. But most losses increase rapidly with the augmentation of the peripheral

velocity, and, though it is common to go as high as 164 feet per second with the peripheral velocity in stationary installations, a suitable value for economical marine-engine operation would be 32.8 to 49.2 feet per second (see example further on).

(2) The velocity in the suction and delivery pipes may be increased to 18 feet per second.

(3) The number of revolutions of the circulating engine should be as high as possible. Here, however, a distinction must be made whether we have to deal with a steam engine that is only temporarily subject to a heavy load, as in torpedo vessels, or with an engine running under such a load for days at a time, as in battleships or express steamers.

(4) It is best to make hub of the impeller of bronze and rivet the blades on.

(5) The circumference of the pump casing should be made to conform to a curve similar to the spiral of Archimedes, and proportioned in such manner that the absolute exit velocity of discharge will be in harmony with the cross-sectional areas and quantities of water.

(6) The pump casing itself should be made as smooth as possible on the inside, by grinding or coating with asphaltum.

(7) The water velocity arising in the spiral must be brought up to the velocity in the delivery pipe through a gradually increasing section of nozzle.

IV.—THE CONDENSER.

The total pressure p_o of air and vapor contained in a condenser is generally composed of two parts:

(1) The pressure l of the air present in the condenser, in pounds per square inch, and

(2) The pressure d of the steam present, in pounds per square inch; so that the following relation will always hold,

$$p_o = l + d.$$

The problem for a good condenser is to produce the smallest possible pressure in order to have for turbines a very large drop in pressure for utilization. It is further desirable that

the mixture to be drawn off should have the lowest possible temperature and consequently possess a small volume so that the air pumps can be kept as small as practicable. For sea-going vessels there is nearly always available an unlimited quantity of circulating water which, in a well-designed condenser, can be conveyed through the same at a proportionally small expenditure of power.

A modern marine condenser should be governed by the following conditions:

(1) The condenser should possess the least possible weight; hence, the importance of attaining the highest practicable efficiency of the cooling surfaces.

(2) The condenser should cause the smallest possible expense. If practicable, a cylindrical form of ample length should be used, so that proportionally few tubes and screw glands are required; at the same time the danger of leaky tubes is therewith reduced.

(3) The inlet for the steam should be, if possible, near the top, so that in descending the steam will be compelled to find its way around numerous tubes. The suction pipe for the wet-air pump should be suitably arranged near the lowest point of the condenser. The best location of suction pipe for the dry-air pump is near and somewhat below the center of the condenser. It would be an advantage, however, to make provision for two suction pipes instead of one, located opposite to one another and joined together in a common pipe.

(4) The flow of the cooling water through the condenser should occur in such a manner that, in all cases, it will enter cold at the bottom and leave hot at the top. The most intensive cooling action and the lowest temperature of the mixture to be drawn off will thus be obtained.

(5) The condenser should receive few connections, so that there will be few flanges to be kept tight in order to prevent leakage of air. It is recommended that provision be made for valve chests having a common connection with the condenser, instead of individual valves. All cock connections with the condenser should be fitted with stuffing boxes. For

the drain connections of the main and auxiliary machinery and piping, as well as the exhaust connections of the individual auxiliaries, baffle plates should be fitted in the condenser for the protection of the tubes. The latter connections are most suitably arranged quite near the top of the condenser, in case no water pockets are formed in the drain connections, so that the steam and hot water, respectively, will be compelled to cool off well before the mixture of condensation flows towards the air pump. At the same time, by so doing a kind of cooling tower is produced that in return acts advantageously on the air, which is very difficult to cool off.

The baffle plates above mentioned are usually made of flat plates perforated with holes. The net area of the holes is from 1.2 to 2 times the area of the pipe.

It is recommended, especially in suction piping from condensers, to make the net area of the holes still larger. In no case should the diameter be less than .196 inch. It is best to make the shape of the baffle plates, as shown in Figs. 19 and 20, if this shape can be adapted to the work.

Fig. 19.

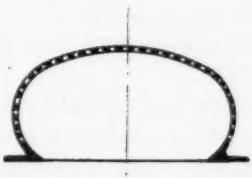


Fig. 20.

The size of the condenser is based on the proposition that the heat withdrawn from the steam must be equal to the heat transferred to the cooling water.

Let t_0 be the temperature of the cooling water at the inlet and t_1 that of the temperature at the outlet, both in degrees C. If, further, n kilograms of cooling water (specific quantity of cooling water) are necessary to condense 1 kilogram of steam, and if r is the heat rejected by 1 kilogram of steam, then the following formula will hold good,

$$r = n(t_1 - t_0),$$

where r is expressed in heat units of the metric system.

Since in most cases the pressure of the condenser is assumed, the heat contained in the steam to be condensed may be regarded as given. The maximum temperature t_0 of the cooling water for this latitude may be taken at 15 degrees centigrade (59 degrees Fahrenheit). Therefore, for a given specific quantity of cooling water (n = about 40 to 50), the temperature at the discharge outlet can be determined.

For a temperature 15 degrees centigrade (59 degrees Fahrenheit), for the cooling water at the inlet the theoretical obtainable pressure would be about 0.032 atmosphere absolute (0.928 inch of mercury).

The size of the condenser surface follows from the following formula:

$$W = F \times k \left(t - \frac{t_1 - t_0}{2} \right);$$

$$F = \frac{W}{k \left(t_2 - \frac{t_1 - t_0}{2} \right)}.$$

In this formula—

W represents the total heat to be withdrawn from the steam in H.U.,

F the cooling surface in square meters,

K the coefficient of heat transmission per square meter per hour and 1 degree centigrade difference in temperature,

t_2 the corresponding temperature of the steam in degrees centigrade of the required vacuum,

t_1 and t_0 the temperature of the cooling water at the inlet and outlet, respectively, in degrees centigrade.

The terms on the right-hand side of the latter equation are all known or can be easily ascertained from what has been previously stated, with the exception of the coefficient of heat transmission k. On this coefficient experiments were made by Messrs. Ser and Josse,* the results of which may be here briefly recapitulated.

*See Josse "Experiments on Surface Condensation for Steam Turbines, Especially for Ships," in a paper read before the Schiffbautechnishe Gesellshaft at the summer meeting, Berlin, June, 1908.

The coefficient of heat transmission k is composed of three quantities:

(1) The coefficient of transference α_1 from the steam to the walls of the condenser tubes,

(2) The thermal conductivity λ through the tube walls,

(3) The coefficient of transference α_2 from the tube walls to the water.

Therefore, according to Mr. Josse, we can write:

$$\frac{1}{k} = \frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2},$$

in which δ represents the thickness of the condenser tubes.

The thermal conductivity λ for various metals is given in Table 2.

Table 2.

Material.	λ for 1 millimeter plate thickness.
Iron, ...	55,000 H.U. per sq. meter per hour.
Copper, ...	30,000
Brass, . . .	90,000 to 100,000
Zinc, ...	100,000
Tin, ...	54,000

The materials coming into consideration for condenser tubes are mostly alloys composed of tin, zinc and copper. The percentage of copper is made sufficiently high (60 to 90 per cent.) that the alloy will play the part of the electro-negative material in sea water, and thus reduce corrosion to a minimum.

The value for α_1 cannot be changed very much, although, according to Mr. Josse, we are able, by increasing the velocity of flow of the circulating water through the condenser tubes from 0.5 to 1.2 meters (1.64 to 3.93 feet) to influence the value α_2 in such a manner that k will be raised from 2,640 to 4,530; hence it follows that we should strive, by all means,

to obtain the highest possible velocity of flow of the cooling water.

The use of retarding strips for increasing the coefficient of heat transmission has not proven a success in marine engineering, since the tubes soon become clogged from sea-weed and small fish.

The thickness of the condenser tubes varies from 0.75 to 1.3 millimeters (0.029 to 0.051 inch). In condensers that are to endure prolonged service, the upper tubes upon which the steam first impinges are made as thick as 1.8 millimeters (0.071 inch). For practical reasons the tubes should not be made longer than 6 meters (19.68 feet). For every 70 to 100 tube diameters in length supporting plates should be fitted 6 to 12 millimeters (0.236 to .472 inch) in thickness.

The pitch t of the tubes, from center to center, for merchant ships and battle ships is made $t = d + 0.354$ inch for torpedo boats. $t = d + 0.236$ inch, in which d represents the external diameter of the condenser tubes in inches.

The tube plates are made of brass, muntz metal or naval brass, 20 to 26 millimeters (0.787 to 1.023 inches) in thickness.

The condenser shell is generally made of plate copper or plate brass. The thickness of the shell, according to Bauer, is about $\frac{D}{350} + 1$ millimeter, if D equals the inside diameter of the condenser.

Around the outside of the condenser itself several tee or channel bars are fitted to prevent collapsing. Lateral bracing must be introduced where necessary to insure safety.

V.—THE LEAKAGE OF AIR INTO THE CONDENSER.

The greatest common evil of all condensing installations is the leakage of air. This leakage is due to two causes:

(1) The water pumped out of the condenser into the feed tank is brought into contact with air of atmospheric pressure. According to Bunsen 1 liter (2.2 pounds) of water at 15 degrees

centigrade (59 degrees Fahrenheit), and 760 millimeters (29.92 inches), barometric pressure, will absorb a quantity of air equal to 0.01795 liter (61.02 cubic inches). This air is continually supplied to the condenser so long as the cycle through which the water passes remains unchanged.

(2) In the low-pressure region of the condenser a larger quantity of air leaks in through cocks, flange connections, piston-rod stuffing boxes, etc. The amount of this air leakage can be essentially reduced by careful workmanship and erection. What measures of design are to be applied to prevent the leakage of air due to non-tightness has already been explained.

During normal running for fast vessels it can be assumed that the weight of the air leakage amounts to about $\frac{1}{1800}$ of the feed water used; for slow-running vessels this amount will be larger and is increased to $\frac{1}{1500}$ of the quantity of the feed water.

From these figures it can be seen that the air leakage into the condenser during the cycle of the water amounts to about $\frac{1}{20}$ of the total quantity of the air. From this we can perceive what extraordinary influeuce good workmanship and good design possesses.

A rough calculation will show that as much air will enter the condenser through a single leaky 10-millimeter (0.393-inch) cock as will be brought in through the cyle of the water during the same time.

The leakage of air into the condenser due to the cycle of the water could be reduced by about one-half if the present feed-water heaters (Normand type, etc.) as now installed were replaced by heaters that would heat the feed water in the feed tank.

VI.—THE AIR PUMP.

The sizes of the air pumps for a marine installation are determined from the values given in the last chapter.

In turbine installations separate wet-air and dry-air pumps are generally installed. A more appropriate term for "wet-

air pump" would be the expression "condensed-water pump."

The purpose of the wet-air pump is solely to draw the condensed water from the condenser and deliver it into the hot-well. As such the pump is generally given a proportionally large cylinder, since, with regard to the slight water pressure available, it becomes compulsory to use only moderate piston speeds. The maximum average piston speed is about 0.5 meter (1.64 feet) per second for small ships, 0.4 to 0.7 meter (1.31 to 2.29 feet) per second for large ships.

With these piston speeds the volumetric efficiency can be assumed at 0.8.

That with respect to the flow of the water and port areas the same conditions apply as explained in the beginning hardly deserves further discussion. Here, also, the valves are again the cause of large losses in work. The three following kinds of valves are in general use:

(a) Beldam valve, Fig. 21;

(b) Kinghorn valve, Fig. 22;

(c) Rubber-flap valve, Fig. 23.

The valves are given a lift of 8 to 12 millimeters (0.315 to 0.472 inch). Therefore, the water in flowing through the valve is compelled to assume a proportionally high radial velocity. In calculations these velocities are taken at 6 to 10 meters (19.68 to 32.8 feet) per second. However, in connection with this, the following deserves consideration:

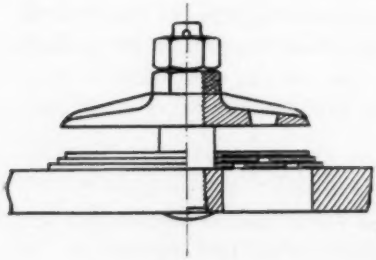


Fig. 21.

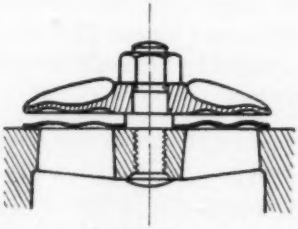


Fig. 22.

Since the pressure prevailing in the condenser amounts to about 0.1 atmosphere, there would thus become available as a suction head a column of water of about 1 meter (3.28 feet). This pressure is further reduced by the air contained in the water and by the steam pressure corresponding to the temperature of the water. Therefore, practically speaking, there is no longer any suction head in general. Consequently, it is an absolute condition that the water flow towards the pump, and the velocity arising in the suction valves will depend solely on the height of the water level in the suction pipe. If the available water column, for example, over the pump valves is equal to 1.5 meters (4.92 feet) and the resistance factor or coefficient of friction ζ of the valve is 1.2, then the only possible velocity in a radical direction is

$$v' = \sqrt{\frac{2gh}{1 + 1.2}} = \sqrt{\frac{2 \times 9.81 \times 1.5}{2.2}} = 3.65 \text{ meters (11.97 feet) per second.}$$

From this it is seen how extremely important it is to place the air pump as low and the condenser as high as possible. Hence, in small ships with little head room, it can readily occur that, through carelessness in designing, the lower part of the condenser becomes submerged and is thus lost for the cooling effect.

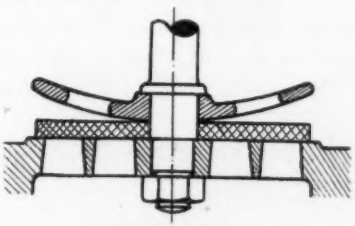


Fig. 23.

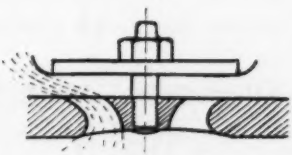


Fig. 24.

For the purpose of giving the valves as favorable an efficiency as possible it is recommended that they be given the form shown in Fig. 24. In a valve of this kind the water

would already receive a gradual acceleration in the valve, and by this a steadier flow of the water would be produced.

The dry-air pumps found in use today are, in general, of the Edwards type, and, on account of their high rate of revolutions, are driven by the piston-rod extension of a steam engine.

Quite recently excellent results have been obtained with centrifugal air pumps of the Westinghouse-Leblanc type.

However, we shall confine ourselves to the first type of pump the design of which may here be assumed to be known. In this pump, above all things, care must be taken that there is always sufficient water over the piston so as to avoid a clearance space on the pressure side. This injection water, which at the same time effects tightness, must be as cold as possible so as to avoid re-evaporation during the downward stroke of the piston. The afflux of the water must be liberally proportioned, since, by the rapid reciprocating motion of the piston, as well as by the impact near the top center, and by the compression of the air, the water becomes rapidly heated and thus the danger arises that re-evaporation will occur during the downward stroke of the piston. The valves will first begin to open when the pressure of the compressed air has increased, at least, so far that it will be in equilibrium with the back pressure of the atmospheric air, plus the weight of the valves. This will occur first, then, when the piston has completed 80 to 86 per cent. of its upward stroke. The valves will then open very rapidly and the air will attain a very high velocity during the outflow.

VII.—GENERAL ARRANGEMENT OF CONDENSING INSTALLATIONS.

If we consider the turbine space of a vessel of the present day, then the first impression obtained is that the arrangement of the machinery demands a large floor space, notwithstanding, however, that the head room is badly utilized (see Figs. 25 and 26). The location of the floors are very high, espec-

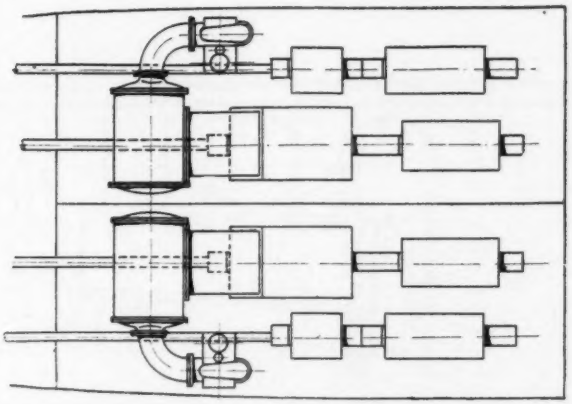


Fig. 25.

ially in the Parson's installation, and the engineer frequently does not have proper supervision over the auxiliary machinery. In this respect the single-shaft turbine arrangements have somewhat the advantage. In the latter the condenser can be arranged directly over the low-pressure turbine as suggested by Dr. Wagner, of Stettin. In Figs. 27 to 30 is shown an arrangement for a torpedo boat, and in Fig. 31 one for an express steamer or large war ship.

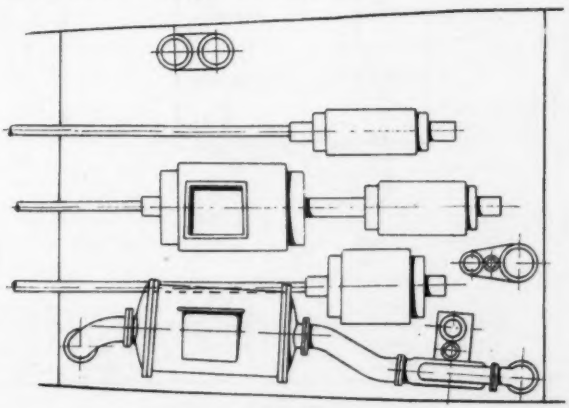


Fig. 26.

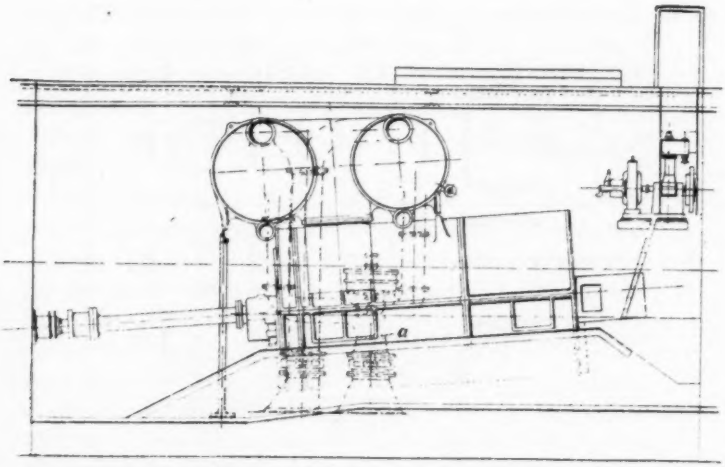


Fig. 27.—AFTER ENGINE ROOM (FORWARD ENGINE ROOM SYMMETRICAL).

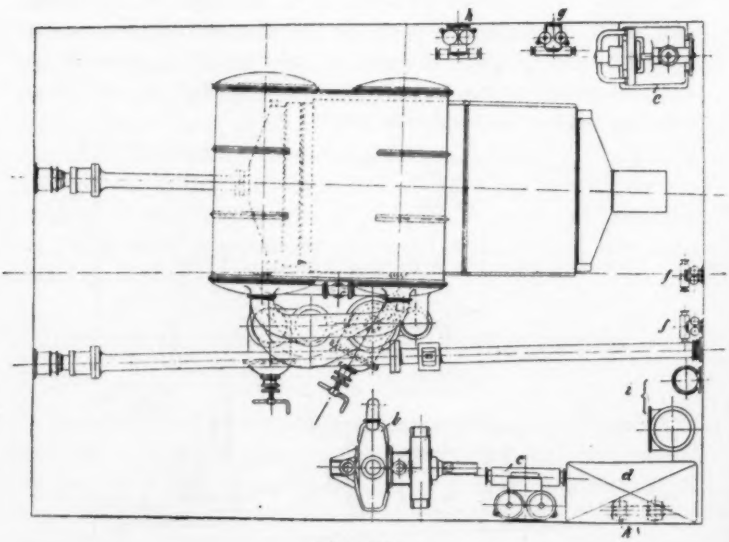


Fig. 28.

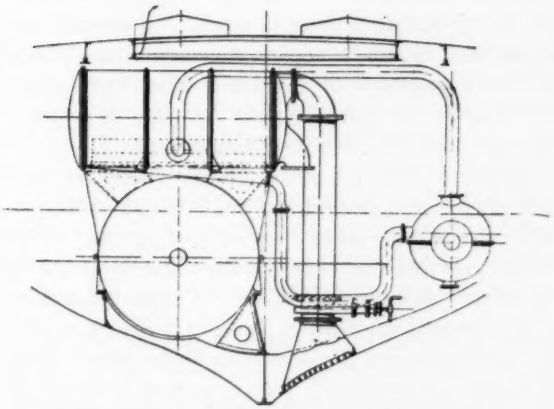


Fig. 29.—For Overhauling Condenser.

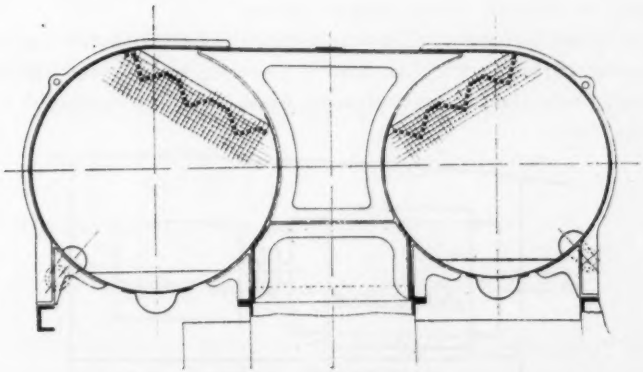


Fig. 30.—Section Through the Condensers.

List of auxiliaries and apparatus.

a, Circulating pump.
b, Combined wet- and dry-air pump.
c, Main feed pump.
d, Feed tank.
e, Electric generator.
f, Oil pumps.
g, Fresh-water pump.
h, Bilge pump.
i, Evaporator and distiller.
k, Evaporator pump.

The advantages of this arrangement are the following:

(1) Saving in space. Tactically the space saved can be utilized for the strengthening of the coal protection or for the stowage of ammunition. A larger space is also more agreeable for the supervision of the entire machinery installation.

(2) Saving in weight. Mainly by the dropping of the exhaust pipe and the condenser foundations.

(3) Improved inlet for steam to the condenser, hence, the same conditions of condensation prevail everywhere. Moreover, the tubes can be more frequently supported by plates.

(4) The width of the tube banks, presenting the greatest cooling effect (greatest coefficient of heat transmission) is larger in a divided condenser than in a single circular condenser. Furthermore, the steam can better penetrate through the individual condensers. Hence, as may be easily foreseen, in such a condenser the total cooling surface can be reduced, and by this means a further saving in weight.

(5) High location of the condenser. In this way a high head is obtained for the wet-air pump.

A disadvantage of this arrangement is the dismantling in case of turbine accidents, which, however, in turbines having large blade clearances and with good attendance should be very rare.

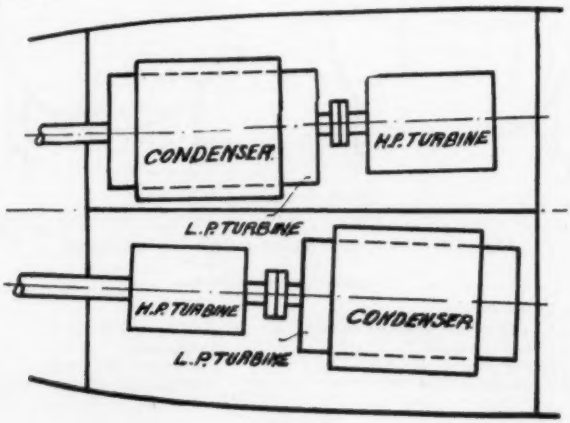


Fig. 31.

Still more compact and lighter can the condensing installation be made if, instead of the dry-air pump a turbine-driven centrifugal air pump be installed, shown in Figs. 27 to 29, as recently brought out by the Allgemeine Electricitäts Gesellschaft, and if instead of a circulating pump for handling the cooling water a turbine-driven propeller equipped with guide apparatus or an axial-reaction turbine be used.

VIII.—CALCULATION OF A CONDENSING INSTALLATION FOR A TORPEDO BOAT WITH TWO SHAFTS AND SINGLE-SHAFT TURBINES FOR A STEAM CONSUMPTION OF 100,000 KILOGRAMS (220,460 POUNDS) PER HOUR.

Each turbine consumes 50,000 (110,230 pounds) kilograms of steam per hour. A vacuum of 90 per cent. is to be attained as a certainty. Let the highest initial temperature of the cooling water be 15 degrees centigrade (59 degrees Fahrenheit), and the quantity of cooling water 45 times the weight of the steam condensed, therefore,

$Q_s = 45 \times 50{,}000 = 2{,}250{,}000$ liters (4,960,350 pounds) per hour.

Then, the quantity of cooling water per second will be

$$Q = \frac{2{,}250{,}000}{60 \times 60} = 625 \text{ liters (1,373.8 pounds).}$$

The quantity of steam per second amounts to

$$D = \frac{50{,}000}{60 \times 60} = 13.9 \text{ kilograms (30.58 pounds).}$$

With the aid of steam tables the final temperature of the cooling water computed from the formula—

$$D \times \lambda = Q\,(t_1 - t_0)$$

is,

$$t_1 = \frac{D \times \lambda + Q \times t_0}{Q} = \frac{13.9\,(620.4 - 45.65) + 625 \times 15}{625}$$

$t_1 = 27.7$ degrees centigrade (81.36 degrees Fahrenheit).

A temperature of about 28 degrees (82.4 degrees Fahrenheit) corresponds to a pressure of

0.037 atmosphere absolute,

therefore, the air may attain a pressure of

$$0.1 - 0.037 = 0.063 \text{ atmosphere absolute.}$$

Assuming, first of all, that the velocity of the cooling water in the tubes amounts to 2 meters (6.56 feet) per second, and, to be safe, let the coefficient of heat transmission be

$$R = 4{,}500,$$

then, the cooling surface will be obtained from the formula—

$$F = \frac{W}{k\left(t_2 - \frac{t_1 - t_0}{2}\right)}$$

$$F = \frac{50{,}000 \times 574.75}{4{,}500\left(45.6 - \frac{28 - 15}{2}\right)}$$

$F = 163.5$ square meters (1,760 square feet).

This would be the amount of cooling surface required if it were not for the detrimental influence of the air. The condensing installations built up to the present time have a cooling surface of 0.008 to 0.009 square meter per kilogram (0.03914 to 0.044 square foot per pound) of steam per hour. According to this the cooling surface would have to be at least twice as large as calculated in the foregoing. In this, however, we must not forget that in the piping of the circulating pump and by the flow of the water through the condenser such considerable resistances always arose that the quantity of cooling water supplied was far below that calculated. A cooling surface of 340 square meters (3,659.76 square feet) would surely be ample.

Let the available length between the tube plates be 2,000 millimeters (6.56 feet). Choosing 16 millimeters ($\frac{5}{8}$ inch) as the inside diameter of the tubes and the thickness as 1 millimeter (0.0393 inch), then the pitch of the tubes will be

$$t = d + 6 = 18 + 6 = 24 \text{ millimeters (0.945 inch).}$$

The arrangement of the condenser is to be as shown in Fig. 30. According to this we get two condensers of 170 square meters (1,829.8 square feet) of cooling surface each. The number of tubes in each of these condensers will be

$$n = \frac{170}{2 \times 0.018 \times \pi} = 1{,}503.$$

The necessary tube-sheet surface without regard to stays, baffle plates, etc., is

$$f = n \frac{t^2}{2} \sqrt{3} = 1{,}503 \times \frac{0.024^2}{2} \times 1.732 = 0.75 \text{ square meter}$$

(8.073 square feet).

Actually, the surface will be about 1.15 times as large, so that there will be obtained as the inside diameter of each condenser

$$l = \sqrt{\frac{1.15 \times 4 \times f}{\pi}} = \sqrt{\frac{1.15 \times 4 \times 0.75}{\pi}} = 1.048 \text{ meters}$$

(3.44 feet).

The velocity of the water in the tubes is determined as follows:

The quantity of cooling water per second is

$$Q = 625 \text{ liters (1,373.8 pounds).}$$

The condensers are so constructed that the cooling water is conveyed along the lower portion in one direction and returned in the upper portion in an opposite direction. Then, the velocity through the tubes will be

$$v = \frac{\frac{625}{2}}{\frac{1{,}503}{2} \times \frac{0.16^2 \pi}{4}} = 2.08 \text{ meters (6.82 feet) per second.}$$

This value agrees very well with the one assumed.

Let the velocities of the water in the suction and delivery pipes of the circulating pump be taken at 5.5 meters (18.04 feet) per second. Therefore the inside diameter of the pipe will be

$$d_s = \sqrt{\frac{4 \times Q}{\pi \times v_s}} = \sqrt{\frac{4 \times 625}{\pi \times 55}} = 3.8 \text{ decimeters (14.96 inches).}$$

Having regard to the high velocity of the water in the pipes, choose the area of the suction grating (ahead grating) somewhat larger than usual, equal to 2.2 times the area of the pipe. The velocity through the grating will then be 2.5 meters (8.2 feet) per second. Then the net area through the grating will be

$$F_g = \frac{625}{25} = 25 \text{ square decimeters (387.5 square inches).}$$

The useful work of the circulating pump is made up of two parts:

(*a*) The work of acceleration, *i. e.*, the work which is necessary to impart to the water the velocity in the delivery pipe. With this velocity the water is discharged overboard. The work to be performed for this velocity is

$$A' = M\frac{v_s^2}{2} = \frac{625}{9.81} \times \frac{5.5^2}{2} = 964 \text{ kilogrammeters (6,972.6 foot-pounds) per second.}$$

(*b*) The work to overcome the hydraulic resistance of the piping and condenser. This work amounts to about

$$A'' = 625 \times 2 = 1{,}250 \text{ kilogrammeters (9,041.25 foot-pounds per second.}$$

It being assumed that the total head of resistance amounts to 2 meters (6.56 feet).

The effective work of the pump is

$A = A' + A'' = 964 + 1{,}250 = 2{,}214$ kilogrammeters (16,013.8 foot-pounds) per second,

or,

$$A = \frac{2{,}214}{75} = 29.5 \text{ effective horsepower.}$$

By assuming an efficiency of 85 per cent. for the circulating engine and, to be on the safe side, an efficiency of only 74 per cent for the circulating pump, the indicated work of the pump will be

$$N_i = \frac{29.5}{0.85 \times 0.74} = 47 \text{ indicated horsepower.}$$

The maximum number of revolutions per minute are not to exceed 400. With an absolute initial pressure of 11.5 kilograms per square centimeter (163.56 pounds per square inch) and a back pressure of 1 kilogram per square centimeter (14.22 pounds per square inch) absolute the pump should just attain the specified performance. Let the cut-off be 60 per cent. and the clearance volume 14 per cent. The theoretical mean pressure will then amount to 0.92 of the absolute initial pressure. If the diagram factor be taken at 0.68, then, for the foregoing given cut-off, we get for the mean pressure

$p_i = 0.92 \times 0.68 \times 11.5 - 1 = 6.17$ kilograms per square centimeter (87.75 pounds per square inch) absolute.

By making the ratio of the cylinder diameter to stroke 1.4 to 1, the stroke will be

$$H = \sqrt[3]{\frac{4 \times 60 \times 75 \times N_i}{1.4^2 \pi \times 2 p_i \times n}} = \sqrt[3]{\frac{4 \times 60 \times 75 \times 4{,}700}{1.4^2 \pi \times 2 \times 6.17 \times 400}}$$

$H = 13.9$ centimeters (5.51 inches).

Then, the diameter of the cylinder will be

$D_p = 1.4 \times 140 =$ say 200 millimeters (7.87 inches).

The following ascertained dimensions correspond in general to the usual method of calculating circulating pumps in marine work.

The diameter of the inlet of the impeller is usually made 0.9 the diameter of the pipe, therefore,

$$d_e = 0.9 \times 380 = \text{say } 340 \text{ millimeters } (13.38 \text{ inches}).$$

The external diameter is to be twice the internal diameter of the impeller, consequently

$$d_a = 2d_e = 2 \times 340 = 680 \text{ millimeters } (26.77 \text{ inches}).$$

The peripherial velocities corresponding to these diameters at 400 revolutions are

$$v_e = \frac{d_e \pi \times n}{60} = \frac{0.34 \times \pi \times 400}{60} = 7.12 \text{ meters } (23.35 \text{ feet}) \text{ per second,}$$

and

$$v_a = \frac{d_a \pi \times n}{60} = \frac{0.68 \times \pi \times 400}{60} = 14.24 \text{ meters } (46.7 \text{ feet}).$$

The theoretical delivery head to be performed by the pump is as follows:

(1) The pressure head corresponding to the work of acceleration

$$h_1 = \frac{v_a^2}{2g} = \frac{5.5^2}{2 \times 9.81} = 1.54 \text{ meters } (5.05 \text{ (feet)}.$$

(2) The resistance head in the piping and condenser

$$h_2 = 2 \text{ meters } (6.56 \text{ feet}).$$

(3) The pressure head lost in the pump itself at an efficiency of $\eta = 0.7$.

$h_3 = (h_1 + h_2)\,0.3 = (1.54 + 2)\,0.3 = 1.06$ meters (3.47 feet).

Therefore, the total pressure head is

$h = h_1 + h_2 + h_3 = 1.54 + 2 + 1.06 = 4.6$ meters (17.08 feet).

This pressure head could be attained with an absolute initial pressure of 11.5 atmospheres (163.56 pounds per square

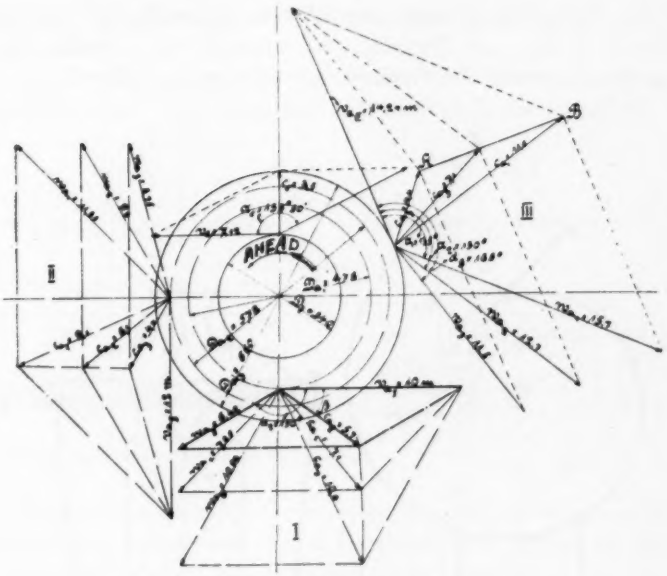


Fig. 32.

inch). Consequently, we see that a circulating pump thus proportioned is sufficiently large.

The radial entering velocity has been assumed at

$$C_r = 3.5 \text{ meters (11.48 feet) per second.}$$

Hence follows 153 degrees 50 minutes for the angle a_r (Fig 32).

From the formula

$$h = \frac{v_a^2 + v_a W_a \cos a_a}{g}$$

follows the product

$$W_a \cos a_a = \frac{hg - v_a^2}{v_a} = \frac{4.6 \times 9.81 - 14.24^2}{14.24};$$

$$W_a \cos a_a = -11.08 \text{ meters (36.34 feet) per second.}$$

The negative value of this expression signifies that the angle a_a lies in the second quadrant.

In Fig. 32 have been plotted the values W_a for various angles of $a_a = 90$ degrees, 120 degrees, etc. Hence, by graphical means, the absolute exit velocity C_a is found.

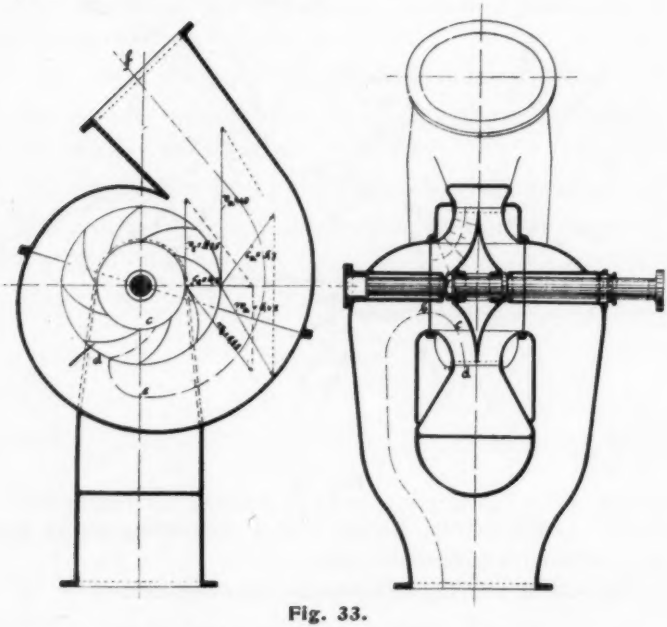


Fig. 33.

If the terminal points of the absolute exit velocities are joined by a line AB, there will be obtained the geometrical loci for any exit velocity within the specified limits.

However, the values for C_a are not satisfactory, especially for the specified peripherial velocity of 14.24 meters (46.7 feet) per second.

Hence, let the following course be pursued:

In the formula

$$h = \frac{V_a^2 + V_a W_a \cos a_a}{g}$$

the external peripherial velocity V_a is given serially the values 10, 12 and 14.24 meters (32.8, 39.36 and 46.7 feet) per

second. The product $W_a \cos \alpha_a$ of this formula can be seen in Table 3.

In a similar manner the proper values of W_a for the various angles α_a have been calculated and entered in the table.

Table 3.

v_b meters per second.	$W_a \cos \alpha_a$.	W_a in meters per second for $\alpha_a =$				
		90°	120°	135°	150°	165°
10.00 (32.8)	— 5.400	∝	10.98 (36.1)	7.85 (25.75)	6.42 (21.05)	6.15 (20.17)
12.00 (39.36)	— 8.247	∝	16.70 (54.77)	11.90 (39.03)	9.75 (31.98)	8.65 (28.37)
14.24 (46.7)	— 10.080	∝	21.50 (70.52)	15.40 (50.05)	12.60 (41.33)	11.20 (36.73)

NOTE.—Figures in parentheses are in feet per second.

If 400 revolutions per minute be retained for the circulating pump then the external diameter of the impeller for the various peripherial velocities will be as follows:

$v_b = 10$ meters (32.8 feet) per second, $d_b = 478$ millimeters (18.81 inches).
$v_b = 12$ meters (39.36 feet) per second, $d_b = 572$ millimeters (22.52 inches).
$v_b = 14.24$ meters (46.7 feet) per second, $d_b = 680$ millimeters (26.77 inches).

These values have been plotted in Fig. 32 and then C_a ascertained. From this presentation the following conclusions can be drawn:

(1) The smaller V_a is chosen the larger will be the tangential component of C_a.

(2) At low peripheral velocities a smaller diameter, and, on account of the smaller friction, a better pump will be obtained than at high peripheral velocities.

These principles, however, only hold good for pumps having such low heads as occur in marine engineering.

For the final construction, therefore, choose 478 millimeters (18.81 inches) as the external diameter of the impeller. For the absolute exit velocity

take $C_a = 5.5$ meters (18.04 feet) per second, and
then $a_a = 150$ degrees (see Fig. 33).

The internal diameter of the impeller is determined as follows:

If the axial entering velocity be chosen at 5.5 meters (18.04 feet) per second, and the diameter of the hub of the impeller be assumed at 95 millimeters (3.74 inches), then the circular-inlet area must be

$$F = \frac{625}{2 \times 55} + \frac{0.95^2 \pi}{4} = 6.36 \text{ square decimeters (98.58 square inches).}$$

This surface corresponds to an internal diameter of

$d_e = 294$ millimeters (11.57 inches) for the impeller.

On account of the great change in direction which the water suffers in the lower part of the pump, take

$d_e = 300$ millimeters (11.81 inches).

The final peripheral velocity at the internal diameter of the impeller will be

$$v_e = \frac{d_e \pi n}{60} = \frac{0.3 \times \pi \times 400}{60} = 6.28 \text{ meters (20.6 feet) per second.}$$

The radial velocity through the impeller may be chosen at 4 meters (13.12 feet) per second, then the entering angle of the blade will be

$a_e = 147$ degrees 30 minutes.

Now, choosing further, the number of blades $Z = 8$ and the thickness $e = 4$ millimeters (0.157 inch), then there will be obained as the width of the impeller at the inlet

$$b_e = \frac{Q}{C_e \left(2 \pi \lambda_e - Z \frac{e}{\sin a_e}\right)} = \frac{625}{40 \left(2 \times \pi \times 1.5 - 8 \frac{0.04}{\sin 147^\circ 30'}\right)}$$

$b_e = 1.83$ decimeters (7.2 inches).

Therefore, the width on each side of the inlet will be

$$\frac{b_e}{2} = \frac{183}{2} = 91.5 \text{ millimeters (3.6 inches).}$$

The width of the impeller at the outlet will be

$$b_a = \frac{Q}{C_a \sin \beta \left(2\pi v_a - Z \frac{e}{\sin a_a} \right)}$$

$$= \frac{625}{55 \sin 35^\circ\ 40' \left(2\pi \times 2.39 - 8 \times \frac{0.04}{\sin 150^\circ} \right)}$$

$$= 140 \text{ millimeters (5.51 inches).}$$

In circulating pumps without guide wheels the greatest care must be bestowed on the shaping of the casing surrounding the impeller. Most casings are given the form shown in Fig. 34 or Fig. 35. The casing shown in Fig. 34 is of constant width around the whole circumference and of variable height h, and is, therefore, easily made. In Fig. 35 the

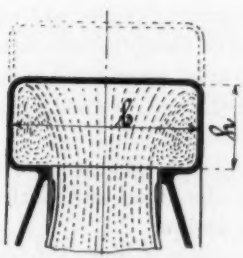


Fig. 34.

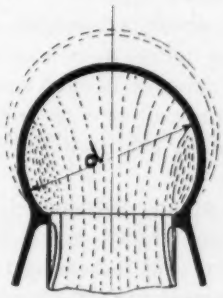


Fig. 35.

diameter d increases with increasing circumference. The cost of manufacture of the core box of such a casing is proportionately high. The distribution of the water is shown in both figures. It can be seen, especially in Fig. 34, that large eddies are formed in the corners nearest the center of the

pump. A more favorable shape, and one that at the same time reduces the cost of manufacture of the pattern, is shown in Fig. 33.

Since the conversion of velocity into pressure is nearly always accompanied by losses it thus appears proper to select one of the parallelograms drawn in Fig. 32, having an absolute exit velocity that comes very near to the velocity in the discharge pipe, or, better still, that is larger by an amount equal to the friction consumed by the walls of the casing.

If a certain exit velocity C_a has been decided upon, then calculate the cross-sectional areas at 6 or 8 different points, which must be available for the passage of the water. Moreover, the radial component of C_a must nearly always be equal to the cross-sectional height h (see Fig. 33).

The deflection of the casing during transition to the pressure pipe shall follow it at an angle B.

A circulating pump designed from this point of view is represented in Fig. 33. The dash-and-dot line represents the path of an element of water on its way from the suction to the pressure pipe.

The developed length of this path is represented as an abscissa in Fig. 36, in which the points designated by the same letters coincide in each case. The absolute velocities due to the corresponding cross-sectional areas are plotted as ordinates. A constant height of the curve warrants a good construction in this case.

The pumps constructed up to the present time often show numerous cracks, especially at the inlet and outlet of the impeller, thus indicating faulty design.

In calculating wet-air pumps it is customary to assume that a fall in head of

$$h = 1.8 \text{ meters } (5.9 \text{ feet}),$$

is available between the lower edge of the condenser and the air-pump valves. This head corresponds to a theoretical velocity of

$$v = \sqrt{2gh} = \sqrt{2 \times 9.81 \times 1.8} = 5.93 \text{ meters (19.45 feet) per second.}$$

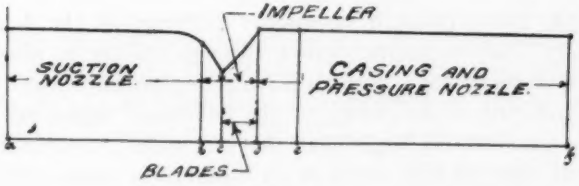


Fig. 36.

With a coefficient of friction $\zeta = 1$ the maximum possible velocity in general through the valves will be

$$v' = \frac{5.93}{1+1} = 2.965 \text{ meters (9.72 feet) per second.}$$

The ratio of the net valve area to the piston area can be made about 1 : 3.8. Now since the column of water behind the piston should not tear asunder, the maximum piston speed must thus be taken into the calculation and assumed that it is $\frac{\pi}{2}$ times as large as the average piston speed. Hence, for the average piston speed, follows

$$c = \frac{v'}{\frac{\pi}{2} \times 3.8} = \frac{2.965}{\frac{\pi}{2} \times 3.8} = \text{say } 0.5 \text{ meter (1.64 feet) per second.}$$

At $n = 40$ double strokes per minute it follows that the stroke

$$S = \frac{30\,c}{n} = \frac{30 \times 0.5}{40} = 0.375 \text{ meter (14.76 inches).}$$

The wet-air pump is to receive two cylinders. Then, for a volumetric efficiency of 0.8 the piston area will be

$$\frac{\pi d^2}{4} = \frac{D}{0.8 \times c} = \frac{13.9}{0.8 \times 5} = 3.47 \text{ square decimeters (53.57 square inches).}$$

Hence, the diameter

$$d = \text{say } 210 \text{ millimeters } (8.26 \text{ inches}).$$

Now, since through occasional priming of the boilers, through leaky condenser tubes, through rolling or pitching of the vessel during which air-pockets are formed in the tubes, the quantity of the water in the condenser sometimes becomes considerably larger than that calculated, thus the diameter of the cylinder must be increased to at least 240 millimeters (9.44 inches).

Frequently there is yet the rule that an air-pump shall be capable of drawing the water of condensation out of both condensers. In this case the diameter of the pump cylinder would have to be made.

$$d' = \text{say } 340 \text{ millimeters } (13.38 \text{ inches}).$$

The pumps as actually built in practice are generally made still larger. This, however, will not be necessary if the condenser be placed sufficiently high.

The main dimensions of the dry-air pump, which is to be of the single-cylinder Edward's type, are found as follows:

The weight of the air entering the condenser and to be drawn off by the pump, as specified in the foregoing amounts to

$$L_1 = \frac{50{,}000}{1{,}800} = 27.8 \text{ kilograms } (61.16 \text{ pounds}) \text{ per hour.}$$

The volume of air corresponding to an air pressure of 0.063 atmosphere absolute is 22.58 cubic meters per kilogram (362.4 cubic feet) per pound. Hence, it follows that the amount of air to be drawn from the condenser per kour is

$$L = 27.8 \times 22.58 = 628 \text{ cubic meters } (22{,}174 \text{ cubic feet}).$$

Assuming that the air pump at maximum performance makes 350 revolutions per minute, then, the volume per stroke to be delivered by the pump

$$l = \frac{628}{350 \times 60} = 0.0299 \text{ cubic meter} = 29.9 \text{ cubic decimeters}$$
(1.055 cubic feet).

If the volumetric efficiency of the pump be assumed at 0.75 and the ratio of the diameter of the cylinder to the stroke is taken as 1.25 to 1, hence, for the diameter of the air-pump cylinder, follows:

$$D_l = \sqrt[3]{\frac{4 \times 1.25 \times l}{\pi \times 0.75}} = \sqrt[3]{\frac{4 \times 1.25 \times 29.9}{\pi \times 0.75}}$$

D_l = say 400 millimeters (15.74 inches).

The stroke will then be

$$S_l = \frac{D_l}{1.25} = \frac{400}{1.25} = 320 \text{ millimeters (12.59 inches).}$$

The mean piston speed will be

$$C_l = \frac{s_l \times n_l}{30} = \frac{0.32 \times 350}{30} = 373 \text{ meters (12.23 feet) per second.}$$

The mean velocity of the air in the suction pipes must be taken proportionally low, since the air only has an opportunity to flow into the pump cylinder when the piston is in the vicinity of the lower dead center. The velocity then arising through the parts is conditioned by the difference between the pressure existing in the condenser and the smaller air pressure produced in the pump cylinder. If no special resistances are present in the suction piping the mean velocity of the air can be taken at 10 meters (32.8 feet) per second.

Dry-air pumps are usually of the vertical type and, in order to obtain speed uniformity, are driven by a reciprocating steam engine arranged with two flywheels at the sides.

DESCRIPTION AND TRIALS OF TORPEDO-BOAT DESTROYERS *PAULDING* AND *DRAYTON.*

By Lieutenant Commander H. C. Dinger, U. S. N.

The Destroyers *Paulding* and *Drayton* are among the ten torpedo-boat destroyers authorized by act of Congress dated May 13, 1908.

Contract for their construction was made with the Bath Ironworks, Ltd., of Bath, Me., and signed September 29, 1908, calling for completion and delivery within two years from that date.

The contracts call for vessels of 742 tons' displacement and 29.5 knots' speed; to be built on plans furnished by the Navy Department as to type, armament and equipment; the machinery to be Parsons turbines capable of propelling the vessel at the specified speed. The weight of machinery was limited to 272.5 tons.

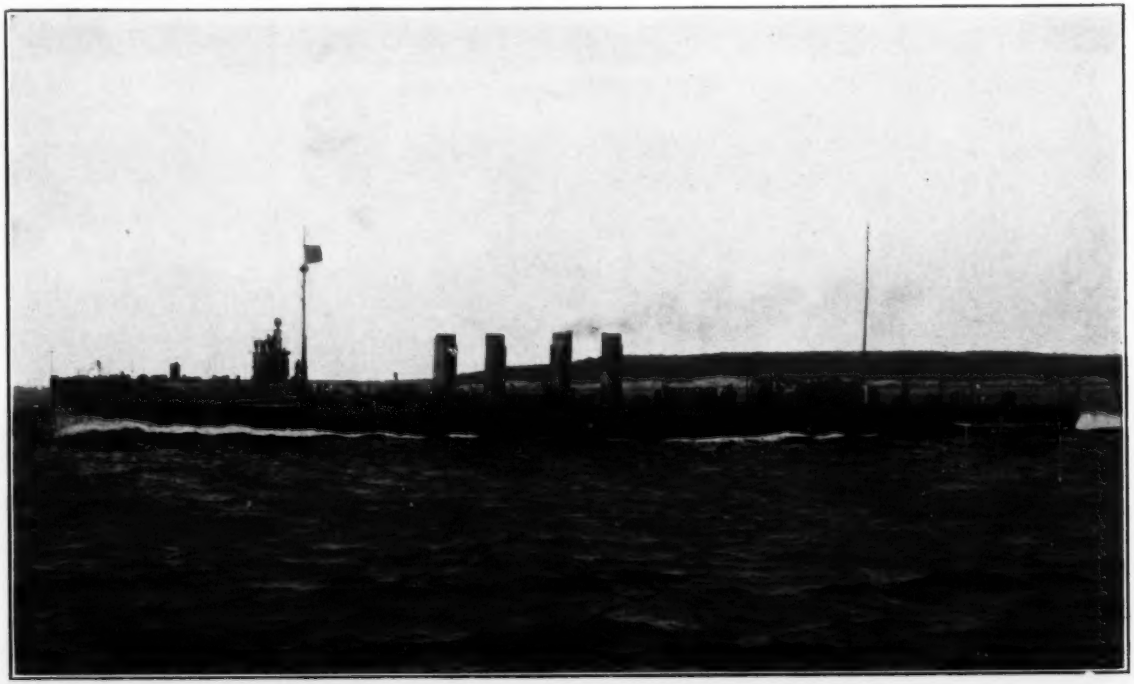
The following trials were required:

A standardization trial over a measured-mile course in deep water, to ascertain the revolutions of the screw propellers corresponding to the measured speeds.

A full-speed trial of four hours' duration, with only the necessary auxiliary machinery in operation and a fireroom air pressure limited to five inches of water, and the steam pressure at the H.P. turbine not to exceed 240 pounds by gauge.

A twelve-hour trial at a nearly constant speed of twenty-five knots, the fuel and steam consumption to be measured.

A twenty-four-hour trial at sixteen knots, during which the



U. S. S. "DRAYTON."

fuel and steam consumption was to be measured. This trial was subsequently reduced to twelve hours.

The following limitations were placed on the steam consumption:

At full speed, 15.5 pounds of water per shaft horsepower per hour.

At 25 knots for twelve hours, 16.8 pounds of water per shaft horsepower per hour.

At 16 knots for twenty-four hours, 25.2 pounds of water per shaft horsepower per hour.

The water in each case is that required for all purposes. The horsepower is that of the propelling turbines only.

The contract price for each vessel was six hundred and forty-four thousand dollars, of which amount two hundred and sixty thousand was for the hull, foundations for and installation of armament, and boat and ordnance outfit complete by the contractors after the last two items shall have been supplied by the Government, and three hundred and eighty-four thousand dollars for the steam machinery under cognizance of the Bureau of Steam Engineering.

GENERAL DESCRIPTION OF THE VESSEL.

HULL.

The hull is built of medium and hard steel. All parts of shell and frame below the water line are galvanized, and all parts are lightened by omitting and cutting out useless material.

There is a main deck and raised forecastle, the latter extending as far aft as frame 41. Under the forecastle are located the quarters for four commissioned officers, a bath room and toilet, the officers' pantry, the ship's galley and a crew's watercloset.

There are four smoke pipes, oval in section, raised hatch coamings for the firerooms, two forced-draft down-takes with concave hoods which can be raised to form deflectors for the wind or can be battened down to keep out the sea or weather.

Abaft the smoke pipes on deck is the engine-room hatch, and abaft of this is the deck house, which includes a crew's watercloset and wash room, a wireless-telegraph room and a hatch to the berth deck.

Under the forecastle forward of frame 12 are the lamp room and a small storeroom. Abaft frame 12 are the officers' quarters. The berth deck extends as far aft as frame 57, and aft from frame 124 to frame 161, or to the rudder post. Between frames 12 and 51 is the crew space in two compartments separated by a watertight bulkhead with a watertight door at frame 28. In the forward compartment are sixteen folding berths, and in the after compartment are forty-four of these berths. One mess table is provided for the forward compartment and two for the after one. The latter compartment is reached from the main deck by a ladder, and the forward compartment is reached through the after one. Forward of the crew's space on the berth deck is the paint and oil storeroom. A smaller storage space is available in the bow. The narrow space in the bow is filled in with concrete of coke and cement. The two forward storerooms are reached from the deck above. The after berth deck is divided as follows: From frame 124 to 138 are the quarters for eighteen men. From frame 138 to 149 are quarters for eight chief petty officers. Metal folding berths are arranged two tiers high. There is one mess table in the after and two in the forward of these compartments. Abaft of the chief petty officers to the stern is a general storeroom. Under these spaces from frames 124–129 are two oil fuel tanks, from 129–138 a handling room and magazine, from 138–149 a fuel-oil tank. Under the storeroom are two trimming tanks of the following capacities: D–9, 8.5 tons; D–10, 4.6 tons. The living spaces are sheathed with No. 27 U. S. G. sheet-iron, painted. These quarters and storerooms are lighted by dead lights and air ports in the ship's sides. There is no artificial ventilation excepting electric fans in living spaces. At forward fireroom bulkhead is a cofferdam between frames 58 and 59. From frame 59 to 51 and under crew space to frame 43 are three oil-storage tanks. Under the crew's space

from 43 to 38 is a handling room and magazine, from frame 38 to 36 a fresh-water tank, from 36 to 28 a wet-provision storeroom and from 28 to 16 a dry-provision and officers' mess storeroom. Next comes the chain locker, and then under the paint and oil room is a 10.3-ton trimming tank, forward of which, in the fore foot, is a 1.3-ton trimming tank.

There are two boiler compartments and one engine compartment. In the engine-room compartment is a reserve-feed tank of 14.7 tons' capacity, between frames 97 and 103.

The capstan is located on the forecastle with its engine secured immediately underneath on the after watertight bulkhead of the lamp room. It is a vertical (two) cylinder steam engine, built by the Hyde Windlass Co., and is controlled by a lever from deck. On the trials this engine hove in 88 fathoms of chain and the anchor in 14 minutes 48 seconds, or at the rate of 5.95 fathoms a minute.

The machinery spaces extend from bulkhead at frame 59 to frame 124. The engine room is included between frames 97 and 124.

Additional fuel-oil storage tanks are located aft of the machinery spaces. The forward tanks are separated from the forward boiler room by a cofferdam 21 inches wide.

The following are the particulars of the hull:

Length between perpendiculars, feet and inches,	289–00
over all, feet and inches,	293–10½
on load water line, feet and inches,	289–00
Breadth, molded, feet and inches,	26–04½
extreme, over fender, feet,	27–00
Depth, molded, main-deck side at frame 83, feet and inches,	16–04¾
Draught, normal, feet and inches,	8–04
Displacement, normal, tons,	742
per inch at designed draught of 8 feet ½ inch, tons,	12
Ratio of length to beam,	11.1

Area of immersed midship section, square feet,	143
L.W.L., plane, square feet,	5,025
wetted surface, square feet,	7,600
Transverse metacenter above C. B., feet,	7.9
Longitudinal metacenter above C. B., feet,	794
Coefficient of fineness, block,	0.41
midship section,	0.66
L.W.L.,	0.67
cylindrical,	0.63
Number of frames, spaced 21 inches,	164
watertight compartments,	40

COMPLEMENT.

One commanding officer; three officers; eighty-two crew—Seamen branch, 26; engine-room branch, 49; special branch, 2; commissary branch, 2; messmen branch, 3.

CAPACITIES OF TANKS.

Compartment.	Gallons.	Tons F.W.	Tons S.W.
Gravity tank,	50.0	....	....
Engine-room feed tank,	....	2.4	....
Reserve-feed tank,	3,335.0	14.7	....
Trimming tank A– 1,	330	1.2	1.3
A– 2,	2,695	10	10.3
D– 9,	2,230	8.3	8.5
D–10,	1,205	4.5	4.6

HEIGHTS.

Above designed L.W.L., which is 8 feet 3½ inches above molded base line:

Top of masts, fore and main, feet,	60–09
Searchlight platform, feet and inches,	30–09¾
Bridge at center, feet and inch,	24–01
Top of forecastle deck at stem, feet and inch,	17–00¼
main deck at stern, feet and inch,	8–00½

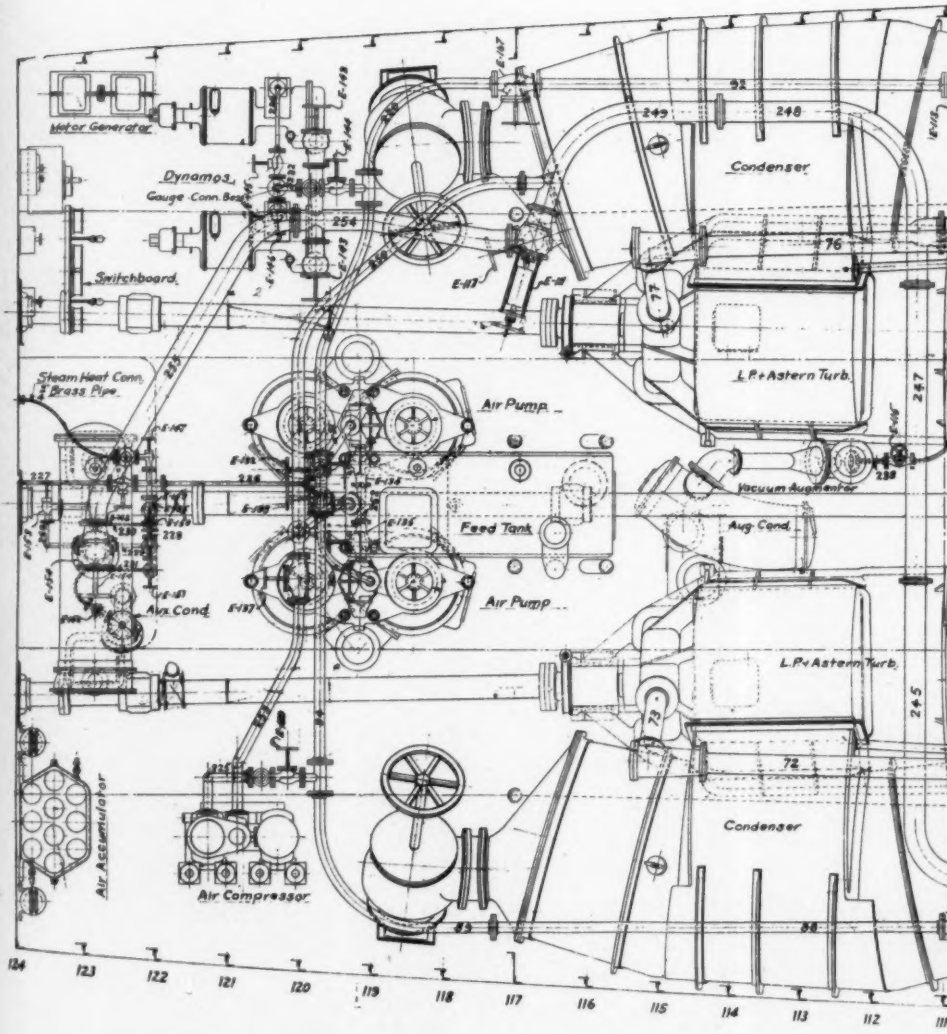
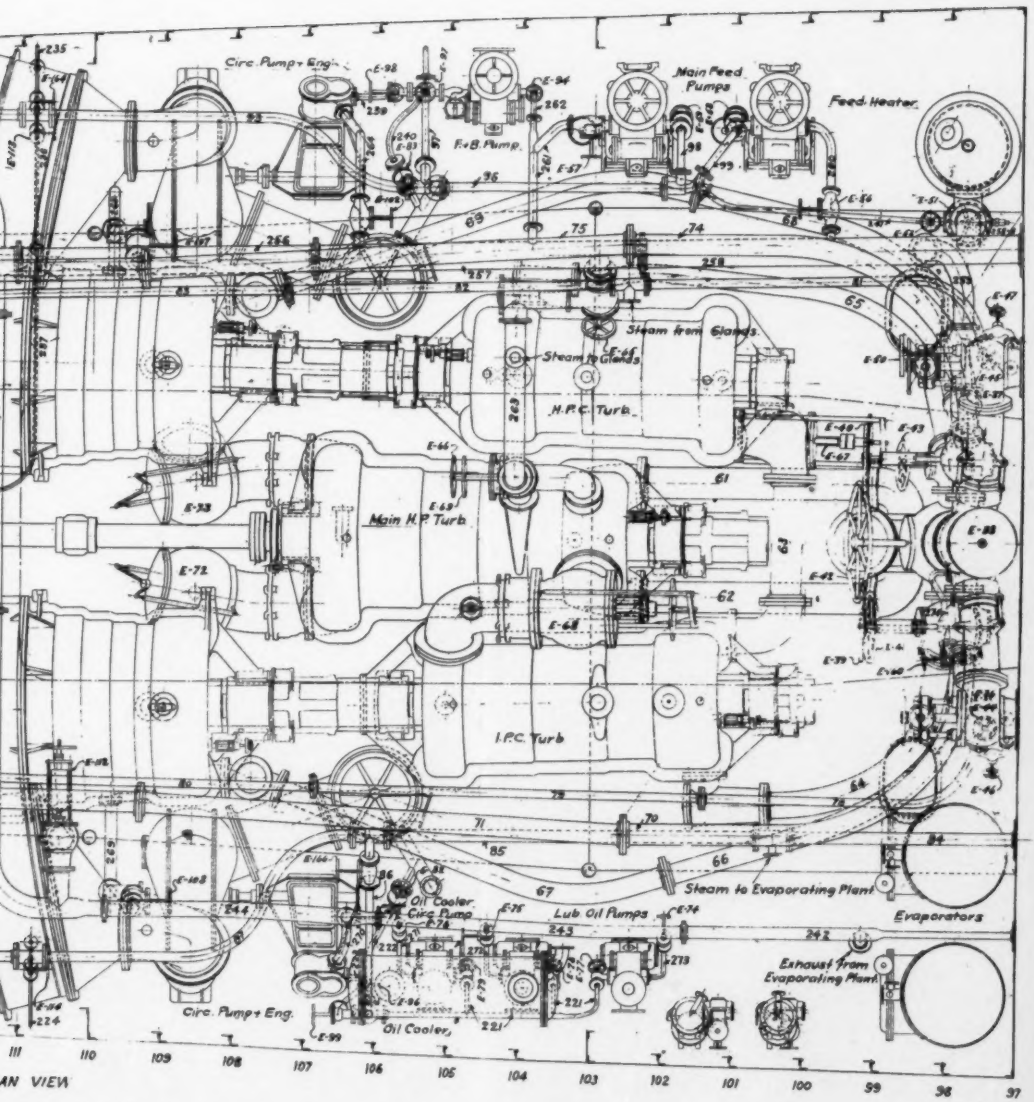


Plate 1.—ARRANGE



EMENT OF ENGINE ROOM.

MACHINERY.

BOILER ROOM.

Admission and egress to and from the boiler spaces is by air locks, one for each boiler room and also by raised hatches, one over each fireroom and one over the rear end of each boiler. Those hatches over the firerooms have spring catches and glass dead lights, the others are secured by bolts and are without lights.

Ample exits are thus provided in case of accident, and the presence of the air lock greatly facilitates supervision.

The boiler rooms are arranged with the oil-settling tanks at the sides, built in with the hull. The tops of tanks are just below the water line, and the tank top forms a platform at sides on which various pumps and auxiliaries are located. This arrangement gives ample space at the sides, and makes it possible to have all the piping accessible.

ENGINE ROOM.

The engine room extends from frames 97 to 124, and is 47½ feet long. The deck is supported by stanchions at intervals. A 36-inch ventilator is fitted at forward end of engine hatch. The hatch is 14 by 9 feet, fitted with spring doors and contains two access ladders and another 36-inch ventilator aft. At the after end of engine room is another hatch 7 by 4 feet, fitted with 36-inch ventilator on port side, and a hatch door and ladder leading to after end of engine room on starboard side.

There are five 8-inch air ports on each side in engine room.

The general lay-out of engine room is shown in Plate I.

TURBINES.

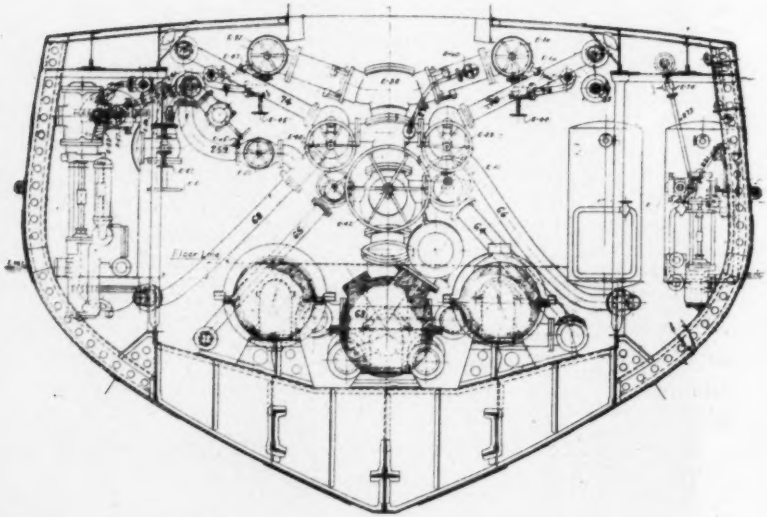
The turbine arrangement is the usual three-shaft Parsons arrangement, with H.P. and I.P. cruising turbines on outboard shafts.

There are five turbine casings, namely: H.P. cruising;

I.P. cruising; main H.P.; starboard L.P. backing and maneuvering; port L.P. backing and maneuvering.

The middle shaft is driven by the main H.P. turbine; each wing shaft by two turbines for the ahead motion and by one for backing. On the port shaft are the H.P. cruising, the L.P. and astern turbines. On the starboard shaft are the I.P.C., the starboard L.P. and astern or backing turbines. Both backing turbines are in the casings of the L.P. turbines.

The working platform is at the forward end of the engine room, and the numerous pipes and valves included in the working gear are located on the forward watertight bulkhead of the engine room, which has been stiffened on its forward side for this purpose. Plate II.



No.	*Service.*	*No.*	*Service.*
63.	H.P.C. exhaust.	70.	Starboard astern steam.
64.	Steam to I.P.C.	74.	Port astern steam.
65.	Steam to H.P.C.	257.	Auxiliary exhaust line.
66.	Starboard ahead maneuvering steam.	259.	Auxiliary exhaust to maneuvering steam.
68.	Port ahead maneuvering steam.		

Plate II.

ARRANGEMENT OF STEAM AND EXHAUST PIPING. SECTION BETWEEN FRAMES 103-104; LOOKING FORWARD.

Composing the group of valves at the working platform are two angle, main stop valves, one on each side; the port valve admitting steam from the two after boilers, the starboard valve admitting steam from the two forward boilers. By converging pipes from these bulkhead stop valves the boiler steam is led into a main governor valve, located amidships and uppermost of the distributing group of valves, from which it passes into a tee immediately underneath, the lower nozzle of which is connected with the main H.P. throttle-valve chamber. The latter debouches into a steam strainer immediately underneath, which latter has two nozzles, leading to the M.H.P. turbine.

On the horizontal arms of the tee are two combination maneuvering valves for admitting and regulating a flow of steam to the L.P. and to the astern turbines on each side of the ship. These maneuvering valves are of the double-beat variety, with their spindles connected by a species of "walking beam" fitted with stiff springs over each valve stem and so arranged that in the process of opening one the other valve is closed. The upper combination valve leads to the astern turbine on its side; the lower to the L.P. (ahead) turbine. By turning the maneuvering-valve wheel right-handed the lower valve is closed, and at the same time the upper one is opened; in other words, steam is shut off the L.P. turbine and admitted to the astern turbine. To maneuver ahead, turn the maneuvering-valve wheel left-handed; to go astern, turn the maneuvering-valve wheel right-handed.

On each side of the M.H.P. throttle-valve chamber is a nozzle to which is secured a throttle valve for admitting steam to the cruising turbines. The H.P.C. throttle is on the port side and connects through a 5-inch steel pipe with the after end of the H.P.C. turbine. The I.P.C. throttle on the starboard side is connected with its turbine by a 7-inch steel pipe at the forward end of the turbine. Both these valves are double-beat valves, operated by hand.

On the pipe connection to the P.L.P. turbine from the port

maneuvering-valve nozzle is a tee and valve connection with the auxiliary exhaust pipe.

Combinations of five, four and three turbines are permissible, either by disconnecting the cruising turbines or by permitting them to revolve in vacuum. For cruising at low speeds the H.P.C. is made the initial turbine by using its throttle. The steam enters the after end of the H.P.C., exhausts across to the forward end of the I.P.C. turbine (through non-return valves), which exhausts in turn into the M.H.P. turbine (through non-return valves), which latter turbine exhausts under all conditions into the two L.P. turbines.

At higher moderate speeds the H.P.C. turbine is put out of use and steam admitted to the I.P.C. turbine through its throttle. At the highest speeds the M.H.P. turbine is made the initial turbine, the cruising turbines being cut out and the main throttle being used.

From the L.P. turbines the exhaust is led through large nozzles of rectangular section connecting the top of each L.P. turbine with its adjacent condenser.

TURBINE DRAINAGE AND GLAND PIPING.

The steam belt, the after end of each turbine and the receiver valves each has a drain connection controlled by a valve operated from grating. The drains for the H.P.C., main H.P. and P.L.P. lead to a drain pocket on after end of P.L.P. casing. The drains for I.P.C. and S.L.P. lead to a drain pocket on S.L.P. casing. From the drain pocket the drainage water discharges into augmenter pipe or air-pump suction.

Steam for glands can be taken either from bleeder pipe or from auxiliary exhaust pipe. The gland system is self-packing, that is, the leak-off of H.P. glands packs the L.P. glands. Cut-out valves are provided for shutting off gland system of each of the cruising turbines.

TURBINE DATA.

Expansion.	.....	Ahead. H.P.C.	Ahead. I.P.C.	Ahead. M.H.P.	Ahead. L.P.	Astern.
1	Diameter of cylinder, inches.	30⅜	30⅜	32⅜	46⅜	31½
1	Length of cylinder, this diameter, inches..	16½	16½	13¼	7	9½
1	Rows of blading	18	16	10	5	8
1	Length of blades, inches	0⅝	0⅞	1⅜	1⅝	0⅞
1	Thickness of caulking sections, pitch of blades...	0⅝	1	1¼	1¼	1½
2	Diameter of cylinder, inches	31	31½	34	48	32½
2	Length of cylinder, this diameter, inches	17½	21⅝	14⅝	7⅛	11½
2	Rows of blading	18	16	10	5	8
2	Length of blades, inches	0½	1½	2	2	1¼
2	Thickness of caulking sections, pitch of blades..	0⅞	1¼	1 5/16	1 5/16	1⅜
3	Diameter of cylinder, inches	31⅜	32	35½	49½	35
3	Length of cylinder, this diameter, inches	17⅜	20½	17	8⅜	14½
3	Rows of blading	18	16	10	5	8
3	Length of blades, inches	0 11/32	1½	2¼	2⅜	2½
3	Thickness of caulking sections, pitch of blades...	0 11/32	1¼	1⅜	1⅜	1⅜
4	Diameter of cylinder, inches	...	...	38	52	35
4	Length of cylinder, this diameter, inches	...	...	18¼	8½	15
4	Rows of blading	...	...	10	5	8
4	Length of blades, inches	...	...	4	4	2½
4	Thickness of caulking sections, pitch of blades...	...	...	1⅜	1½	1⅜
5	Diameter of cylinder, inches.	...	...	...	54	...
5	Length of cylinder, this diameter, inches	...	...	...	8	...
5	Rows of blading	...	...	...	4	...
5	Length of blades, inches	...	...	...	5	...
5	Thickness of caulking sections, pitch of blades...	...	...	...	1 13/32	...
6	Diameter of cylinder, inches	...	...	...	59	...
6	Length of cylinder, this diameter, inches	...	...	...	8½	...
6	Rows of blading	...	...	...	4	...
6	Length of blades, inches	...	...	...	7½	...
6	Thickness of caulking sections, pitch of blades...	...	...	...	1⅜	...
7	Diameter of cylinder, inches	...	...	...	59	...
7	Length of cylinder, this diameter, inches.	...	...	...	8	...
7	Rows of blading	...	...	...	4	...
7	Length of blades, inches	...	...	...	7½	...
7	Thickness of caulking sections, pitch of blades...	...	...	...	2	...
8	Diameter of cylinder, inches	...	...	...	59	...
8	Length of cylinder, this diameter, inches	...	...	...	8	...
8	Rows of blading	...	...	...	4	...
8	Length of blades, inches	...	...	...	7½	...
8	Thickness of caulking sections, pitch of blades...	...	...	...	2	...

Diameter of rotor drums: H.P.C., 30 inches; I.P.C., 29 inches; M.H.P., 30 inches; L.P., 44 inches; astern, 30 inches.

CONDENSERS.

There are two condensers of the Normand type, located one on each side of the engine room, elevated considerably above the turbines, resting in three saddles well up against the ship's

side. They are built of steel plates in three sections with brass tube plates at their ends. The water chests are copper-plate cones, connecting each end of the condenser with the outboard valves. The circulating water is supplied by the main circulating pump. The tubes are of brass of the following composition: 70 copper, 1 tin and 29 zinc, and are expanded into the tube sheets, 1½ inches thick. To allow for expansion of the tubes the condenser and tubes are bent in horizontal planes to a radius of twenty-two feet. The tubes are fixed at each end, and as they expand they move laterally. They are supported at three intermediate points of their lengths by bridges at the bottom of the condenser. At each point of support over these bridges, and between the layers of the tubes, are placed strips of brass one-eighth of an inch thick, three-quarters of an inch wide, and turned up at their ends, where they fit into grooves formed by channel steel riveted to the interior sides of the shell.

There are four passages or lanes formed by omitting tubes to permit the exhaust steam to freely reach the center portions of the condensers.

Baffle plates are fitted along the sides of condenser between the eleventh and twelfth rows of tubes from bottom.

PARTICULARS OF ONE CONDENSER.

Length of shell over tube heads, feet and inches	11–07
Outside diameter of shell, feet and inches	4–05
Thickness of tube sheets, inches, (unbored 00⅞),	0–01⅛
shell plates, inch	0–00½
Number of tubes, each condenser	2,795
Outside diameter of tubes, inch	0–00⅝
Thickness of tubes	No. 18, B.W.G.
Length of tubes, mean, feet and inches	10–06
Cooling surface, outside surface of tubes, square feet	4,800
Area through tubes, square inches	609.59
of circulating water, inlet and outlet, square inches	240.53
through tubes divided by circ. area of inlet, square inches	1.95
Spacing of tubes, between centers, inch	0¾

There are the following connections to each condenser:

One main exhaust, rectangular, area, square feet	12½
auxiliary exhaust, diameter, inches	6
bleeder, diameter, inches	2½

One feed-heater drain, diameter, inches	2¼
Evaporator steam-head drain on port condenser	
Heater drains	
One boiling-out connection, diameter, inch	0¾
safety valve, diameter, inches	2
air-pump suction, diameter, inches	10
feed make-up (port condenser only), diameter, inch	0¾

The following connections are on each water chest:

One main injection on forward chest, diameter, inches	17½
outboard delivery on after chest, diameter, inches	17½
drain, diameter, inch	0½
drain, diameter, inch	0¾
air cock at highest part of water chests (piped), diameter, inch	0¾
thermometer connection	
manhole and cover, diameter, inches	11 × 15
Two hand holes, bottom of shell, diameter, inches	5 × 7

The condensers are stiffened by encircling bands of bulb angles three inches deep, and the shell is built up of three cylindrical sections with riveted seams, chipped and calked, the axes forming a broken line of three segments.

Main circulating pumps, one to each main condenser, are of the centrifugal type with 28-inch runner, driven by an inclined single cylinder, 7 inches by 5 inches, direct-acting engine, with piston valve, and provided with forced lubrication.

The capacity of these pumps is specified as 7,500 gallons per minute. On their suction pipes are interlocking sea and bilge suction valves.

Diameter of runner, inches	28
injection, inches	17½
Number of vanes	7
Diameter of discharge, inches	17½
bilge suction, inches	7
Revolutions of engines, full power	290

The air pumps, of which there are two, of the duplex, single-acting, Blake type, 10 × 22 × 16, are located as low in the vessel as possible, in the after end of the engine room, their working beams in fore-and-aft line.

The augmenter consists of a steam ejector, which takes its suction from the tops of the large suction pipes, between the air pumps and the condensers, and discharges into the augmenter condenser, and through it into a water-seal box in such a way as to force the discharge from it into the channel ways of the main air pumps. By the use of this augmenter the vacuum is increased an inch to an inch and a half below that possible without its use, and to the excellent vacuum obtained by its use is attributed, in large degree, the high speed and economic performance of these vessels.

PARTICULARS OF THE AUGMENTER CONDENSER.

Material of the shell, copper, formed to a curved cylinder slightly enlarged at its middle; thickness of shell, inch	0.1375
Tube sheets and manifold heads are of composition.	
Thickness of tube sheets, inch	00$\frac{7}{8}$
outer heads, inch	00$\frac{1}{4}$
Radius of inner or shortest element of shell, feet and inches	6–03$\frac{1}{4}$
outer or longest element of shell, feet and inches	4–05$\frac{7}{16}$
all tubes, feet and inches	5–05$\frac{1}{2}$
Versine of angle of planes of the two tube heads to one another, multiplied by radius of all tubes (curvature of), inches	14$\frac{1}{8}$
Number of tubes	225
Diameter (outside) of tubes, inch	00$\frac{5}{8}$
Length of tubes, inches	53
Greatest diameter of shell, inches	21
Cooling surface, square feet	139.188

The tubes are not tinned, and are arranged as described for the main condensers.

As to the additional weight of the augmenter and augmenter condenser the total is as follows:

One vacuum augmenter, pounds	145
augmenter condenser, pounds	745
Steam and water piping and valves, pounds	1,143
Total	2,033

OIL-FUEL SYSTEM.

The system employed is the mechanical atomization system. This consists of storage tanks, tank-supply pumps, settling tanks, oil-service pumps, heaters and burners.

From the storage tanks the oil fuel is pumped by two Blake simplex oil-supply pumps to the settling tanks, of which there are four, two alongside of each boiler compartment. These oil-supply pumps are connected so that they can discharge from one storage tank into another, so that they can discharge oil to another vessel, or so that they can pump oil on board from a barge or oil carrier.

From the settling tanks the oil is pumped by means of the oil-service pumps, two in each boiler room. These pumps discharge through oil heaters, strainers, and then to burners. A small hand pump is supplied in each boiler compartment for starting up when there is no steam on vessel.

Storage Tanks.—These tanks are fitted with 4½-inch vents leading to non-return mushroom openings on deck. There is a deck-filling connection, and provision for blowing out tanks by means of steam hose is also provided.

Settling Tanks.—These tanks have 4½-inch vents leading to deck, are each fitted with a gauge glass, and have a small hand pump for pumping overboard water collecting in bottom of tank. Steam-heating coils are fitted in the tanks near suctions.

Oil Heaters.—There are two oil heaters in each boiler compartment. The heating surface of each heater is 131 square feet, there being 149 ⅝ steel tubes 18 B.W.G. thick. The heaters are built in a manner similar to the feed-water heaters, but oil passes three times through the tubes. Live steam is used as the heating agent. The heaters are drained to traps and the discharges are arranged so they can be led to a testing tank to observe whether there is any oil leakage to steam side of heater.

FORCED-DRAFT BLOWERS.

The blower installation on these vessels is somewhat novel. It consists of two blower units, one for each boiler room. Each set consists of a Terry, single-stage, four-nozzle turbine driving a horizontal shaft, upon which are located two single-

inlet and one double-inlet centrifugal cone-wheel fans. Each fan has nine vanes curved back from the direction of rotation. The diameter of fan wheel is thirty-nine inches. The blower units are located on top of the settling tanks in the port after corner of forward and port forward corner of after boiler room. A light air duct leads down from the blower intakes to the inlet for the fans.

Ventilating Duct to Engine Room.—A duct leads from the port side of engine room to the suction of the F. D. blowers, and through this the blowers assist in ventilating the engine room.

OTHER BOILER-ROOM AUXILIARIES.

In addition to the blowers and the oil-fuel system and its accessories each fireroom has a $13 \times 9 \times 14$ Blake simplex auxiliary feed pump and a fire and bilge pump, Blake simplex, $8 \times 5 \times 12$, which has sea and bilge suctions, and discharge overboard and to fire main.

Hose connections are provided for cleaning boiler tubes with either air or steam, the air being supplied to a main, via reducing valve, from the torpedo air compressor.

BOILERS.

There are four boilers of the Normand, return-flame express type, each with a single furnace fitted with eleven oil-fuel burners. Two boilers are in each of the two boiler compartments. To compensate for the squatting of the stern of the ship at high speeds the boilers are pitched forward, or slightly depressed at their forward ends. For draining the lower drums drain cocks are installed at the forward ends of boilers A and C, the bottom-blow valves on the front ends of the two other boilers serving this purpose.

The Normand boilers differ from the other types of boilers used on our oil-burning destroyers in that they have approximately the same baffling arrangement that has so successfully been employed with the Normand coal-burning boiler. In the Thornycroft, White-Foster and Yarrow oil-burning boilers

there is no tube baffling, the products of combustion passing at will from the furnace space through the tube nest to uptakes. In the Normand boiler there is a water wall on the furnace side extending about three-fourths of the way back, and a water wall on the outside extending from the back about three-fourths of the way. About a third of the way from the front of the boiler is placed a steel athwartship baffle plate to direct the flame from the upper portions of the tubes to the lower, which is to say, to prevent the flame from passing horizontally from the rear to the front, and so on up the uptake, without passing among the tubes at their lower portions. This baffle plate is perforated.

This baffling undoubtedly results in some economy, but perhaps the greatest practical advantage is that the side casing is protected against great heat and will therefore not warp or burn out as readily; on the other hand, somewhat harder service is brought upon the back brick lining, since all the gases have to go toward the back of the boiler.

The boilers are fitted with a steam dome at their front ends. On the dome are the main and auxiliary stop valves, both having operating gear from deck.

The steam drum is made in two pieces, with a treble-riveted lap joint on each side. The steam drum has a safety-valve connection with three four-inch valves near back end. In front head there is also a manhole 11 × 15 inches, a main-feed connection, with feed stop and check valves on starboard side, and an auxiliary-feed connection on port side; also a surface blow valve, four gauge cocks and two water gauges fitted with reflex glasses. The surface-blow and feed connections have internal pipes. The lower drums are seamless-drawn steel tubes, reduced in thickness in parts not forming tube sheet. The after ends of water drums are bumped and are fitted with a hand hole for cleaning and examination. The forward ends are also bumped and are fitted with a 14 × 11-inch manhole.

There are two 8-inch down-comer pipes at after end of boiler. At the front are two hollow stays, 3¼ inches outside with 1⅛-inch hole, between the upper and lower drums, and a

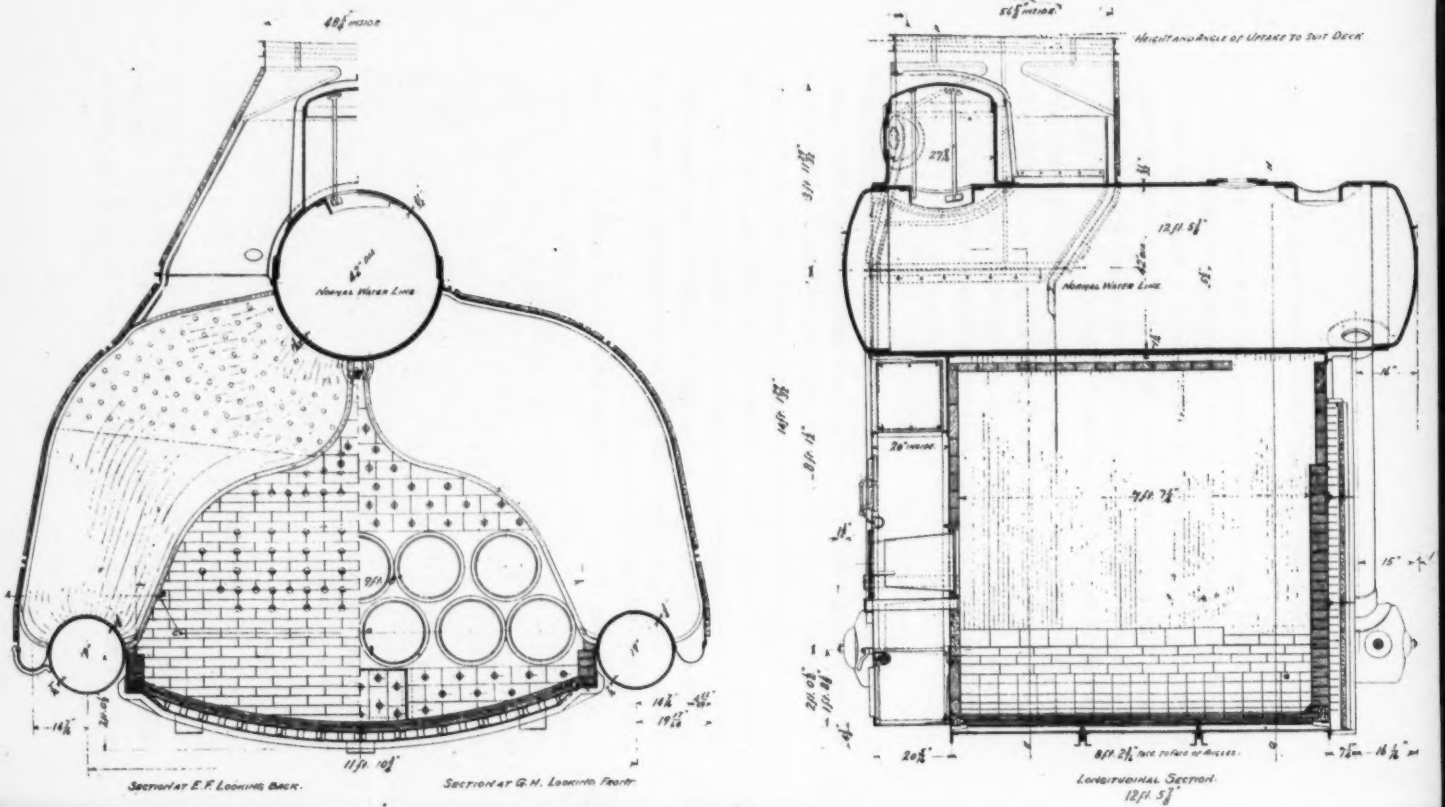

Normal Water Line
Section at E.F. Looking Back.
Section at G.H. Looking Front.
Height and Angle of Uptake to suit Deck
Normal Water Line
Longitudinal Section.

Plate III.—NORMAND BOILER.

Plate III.—NORMAND BOILER.

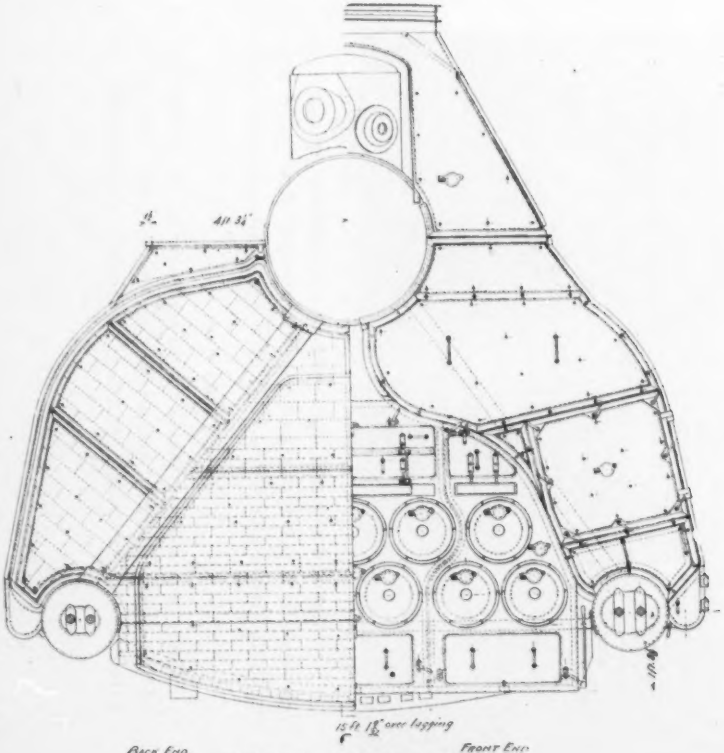


Plate IV.—NORMAND BOILER.

solid stay between the lower drums at each end. Zincs are fitted in the water drums.

The burners, eleven in number, are located in the boiler front. The boiler front has a double casing, with a twenty-inch air space through which the air for combustion is drawn. The front of this casing is fitted with air doors, and the amount of opening can be regulated. The inner front casing has an access door through which entrance into the furnace is effected.

The back of furnace and the sides up to top of water drums are lined with $9 \times 4\frac{1}{2} \times 2\frac{1}{2}$ fire brick, bricks being laid flat so as to have a wall $4\frac{1}{2}$ inches thick. The front is lined with the same size fire brick laid on edge, making a wall $2\frac{1}{2}$ inches thick. The anchor bolts for brick at back and sides only go half way into the brick, and the heads are protected by half the width of a brick. This prevents the heads from burning off. The bottom of furnace is lined with two layers of split bricks laid so as to break joints. Retort cement mixed with powdered fire brick is used for laying bricks.

Around the cylinders of the burners special shapes of fire brick are used. The furnace is thus completely lined with fire brick except the inner walls of tube nest.

The back casing has a three-inch air space, and the bricks at back of furnace are perforated at intervals to allow air from this space to pass into the furnace. In front, over tube nests, the casing is made into doors so that the whole tube nest can be thrown open for examination and cleaning.

The uptakes are built up from the casing and consist of a 11 U. S. S. G. plate, a one-inch air space, a 11 U. S. S. G. plate, 1 inch magnesia, and a 16 U. S. S. G. plate outside.

The steam drum and dome are insulated from uptake by a one-half-inch air space, one-quarter-inch asbestos board, one-inch magnesia and a No. 22 U. S. S. G. galvanized-steel plate.

SMOKE PIPES.

There is a separate smoke pipe for each boiler. The tops of the pipes are about twenty feet above the deck. Guys are fitted for use in rough weather; ordinarily no guys will be

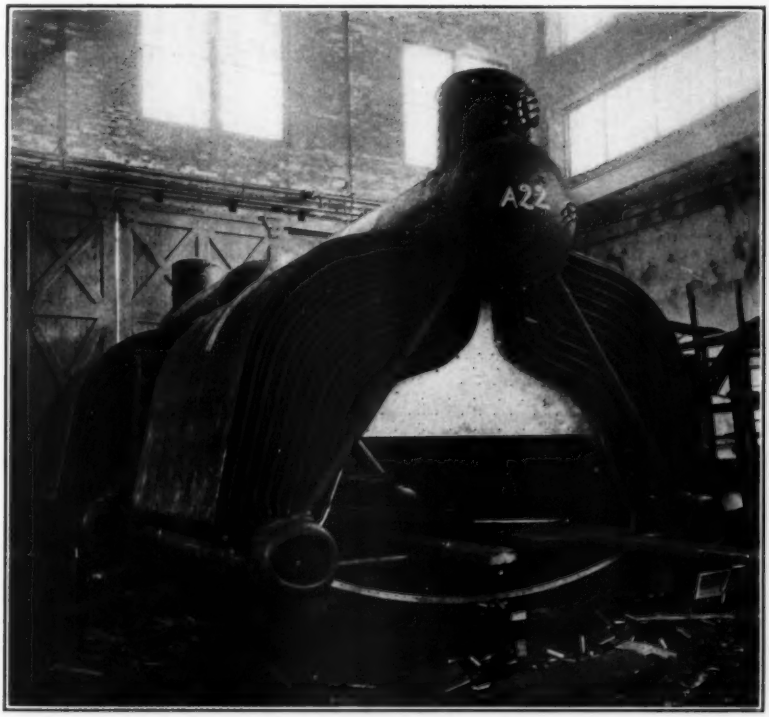


Plate V.—NORMAND OIL-BURNING BOILER.

rigged. There is an inner and an outer pipe with a four-inch air space between. The inner pipe is attached to the uptake, the outer pipe is attached to inner pipe and is open at bottom for entrance of air to air space.

OIL BURNERS.

The oil burners are of the Normand type and employ the Normand tuyere. The details of this burner and tuyere are shown on Plates VI and VII.

The principle of this apparatus is the same as in other systems used on other destroyers of this class, but the details and the method of regulation differ considerably.

PARTICULARS AND DIMENSIONS OF THE BOILERS.

Number of boilers (two in each boiler compartment)	4
Designed working pressure, pounds	260
Hydrostatic test applied, pounds	450
Steam-pressure test applied, pounds	300
Length of boiler over all, feet and inches	12–05$\frac{7}{8}$
Width of boiler over all, feet and inches	15–01$\frac{9}{32}$
Length of furnace, feet and inches	7–09$\frac{1}{2}$
Width of furnace, feet and inches	9–08
Furnace volume, cubic feet	336.1
Number of furnaces	1
burners	11
Heating surface, total area, square feet	19,321
Ratio of furnace volume to heating surface	1 to 14.37
Number of tubes, 1$\frac{1}{2}$ inches diameter	1,834
1$\frac{3}{4}$ inches diameter	118
Thickness of tubes	No. 11, B.W.G.
Kind of tubes	Cold-drawn seamless steel.
Total number of tubes to each boiler	1,952
Length of longest tubes, feet and inches	11–02$\frac{1}{16}$
shortest tubes, feet and inches	8–02$\frac{3}{16}$
Tubes project $\frac{3}{16}$ inch beyond tube sheet and are belled.	
Smoke pipes, number of	4
height above deck, feet and inch	20–00$\frac{7}{8}$
area of section, each, square feet	16.43
Thickness of inner stack	12 B.W.G.
outer stack	16 B.W.G.
Drum, upper, diameter of inside, inches	42
thickness of top sheet, inch	0$\frac{1}{2}$
lower sheet, inches	1$\frac{3}{16}$

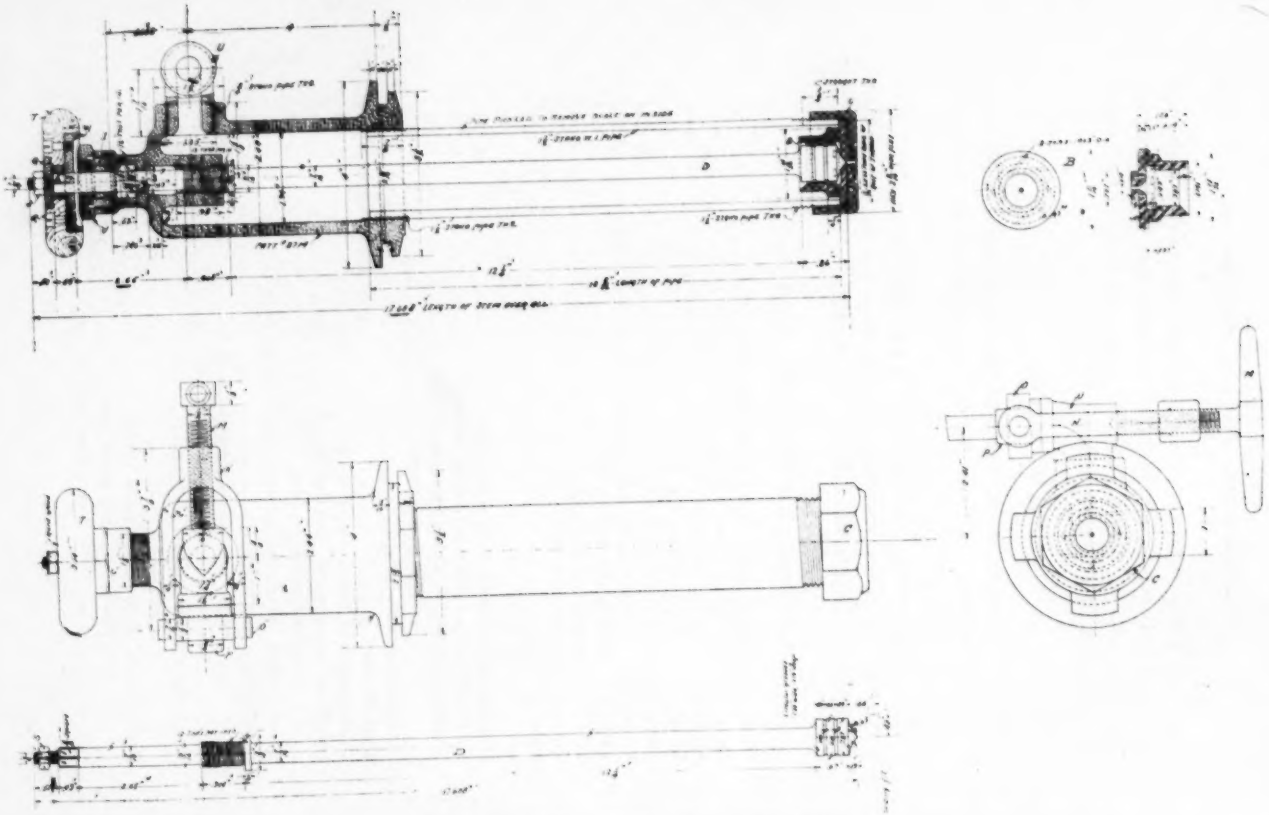


Plate VI.—Oil Burner—Normand System.

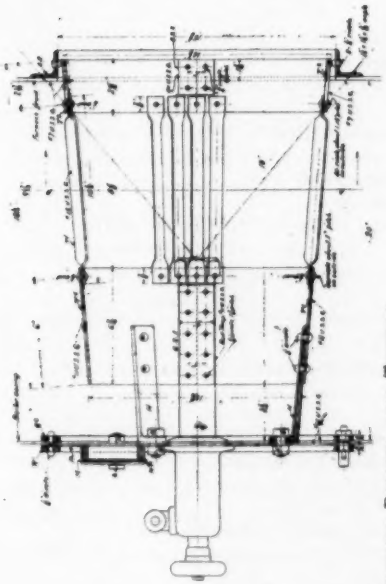


Plate VI.—Oil Burner—Normand System.

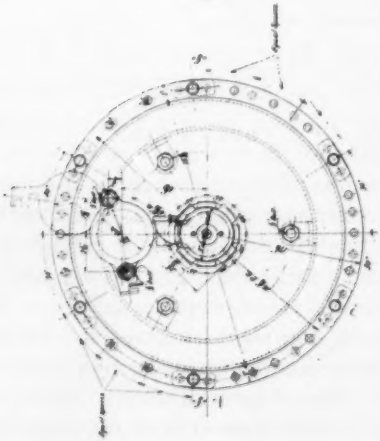


Plate VII.—Normand Burner and Tuyere.

Dome, steam, diameter of inside, inches	$27\frac{5}{16}$
thickness of, inch	$0\frac{5}{8}$
The upper drum heads are bumped to a radius of, inches	42
have thickness of, inch	$0\frac{13}{16}$
The dome head is bumped to a radius of, inches	$27\frac{9}{16}$
has thickness of, inch	$0\frac{5}{8}$
is braced to a stiffening ring of angle steel whose projected diameters are, inches	$15\frac{5}{8}$
The stiffening ring is riveted to the shell and is made of angle steel, inches	$3\times4\times\frac{1}{2}$
Dome braces, diameter of, inch	1
number of	3
Down-comer pipes, diameter of, inside, inches	$7\frac{5}{8}$
outside, inches	8
Tie rods, hollow, front, of mild steel (class B), outside diameter, inches	$3\frac{1}{4}$
Tie rods, hollow, front, of mild steel (class B), inside diameter, inches	$1\frac{1}{2}$
Lower drums, diameter of inside, inches	19
thickness of top sheet, inch	$0\frac{5}{8}$
lower sheet, inch	$0\frac{7}{16}$
Horizontal tie rod between lower drums, solid, diameter of (one of these at each end of boiler), inches	$2\frac{5}{8}$
Weight of one boiler with all appendages dry, without fittings and mountings	50,752 lbs.=$22\frac{14}{32}\frac{7}{16}$ tons.
water and steam at steaming level, 250 pounds gauge pressure,	8,561 pounds=$3\frac{18}{32}\frac{9}{16}$ tons.

Spaces for passage of gases of combustion are provided by omission of tubes, thus forming diagonal alleys through the nest of tubes at each end of the furnace where necessary.

LUBRICATION.

The forced lubrication consists of an oil-cooler circulating pump, two forced-lubrication pumps, an oil cooler, oil-piping drain manifold and drain tank. The pumps and oil cooler are located on starboard side of engine room abaft evaporating plant and forward of circulating pump.

Lubrication of the main turbine, spindle bearings and thrust bearings is accomplished by oil forced through them by the pumps. The oil passes through the cooler on its way to the bearings and is collected from the latter by a system of drain pipes which lead into a tank of one hundred and fifty

gallons capacity, located amidships on the starboard side, just abaft the main high-pressure turbine. This tank is so low in the ship as to insure a rapid flow, visible through an oblong pane of glass in the top of a manifold collector at the tank. At the discharge nozzle of each pipe into this manifold is a thermometer. These thermometers are together in a fore and aft row, and inclined so that they may be read by the light of an electric lamp conveniently placed, and by an observer standing on the engine-room platform. During the trials a series of observations was made by an observer at close hand, and the results recorded on a bulletin board with chalk, so as to be seen more readily from the working platform. Circulating water is supplied by the oil-cooler circulating pump and from a pipe connection from the main circulating-pump discharge pipe to the starboard condenser. The connection to main circulating pump is ordinarily used.

The line-shaft bearings are not connected with the forced-lubrication system, but have ring oilers. These bearings are attended from the after handling room, and from the shaft pits reached from after crew's compartment.

The main circulating-pump engines also have forced lubrication, connections being made so that the main lubricating system can supply oil to these engines.

The oil cooler is of the surface-condenser type, the tubes of which are expanded and sweated into the two tube sheets. By division on the heads the oil is made to pass through the tubes four times. The water enters at one end of shell, and by means of six baffles is made to pass six times across the tubes. The circulating water is taken from the oil-cooler circulating pump and from the discharge of the starboard main circulating pump. The shell of the cooler is a brass tube flanged at each end. The oil heads are of cast composition, and, like the tubes, of the following mixture: Copper, 70 per cent.; tin, 1 per cent.; zinc, 29 per cent. The shell tube heads and water heads are bolted together by two stiffening rings of class B steel, and through bolts and collar bolts.

PARTICULARS OF THE OIL COOLER.

Diameter of shell, inside, inches	17½
Thickness of shell, inch	00½
Length of shell, feet and inches	4–05½
Tube sheets, thickness of, inch	00⅝
length, feet and inches	4–07
diameter of, and of flanges, inches	21¾
Oil heads, diameter of, inches	21¾
thickness of, inch	00½
Circulating-water nozzles on shell, diameter of hole, inches	02½
Tubes, pitch ¾ inch, number of	280
outside diameter of, inch	00⅝
Area through each nest, square inches	15.27
of cooling surface, square feet	202.33

FEED PUMPS.

Main feed pumps, two in number, 13 × 9 × 16 Blake simplex, are located in the port forward corner of the engine room. They have a nine-inch combined suction from the feed tank. The bottom of feed tank is about six inches above the suction valves of pump. The pumps discharge into a common discharge pipe five inches in diameter, which discharges through or by-passes the feed-water heater. The suction pipe of each individual pump is 4½ inches. No grease extractor is fitted.

The auxiliary feed pumps are the same size as the main, and one is located in each fireroom.

The feed and filter tank is located immediately over the air pumps, and is suspended from the main-deck beams. It is built of light galvanized sheet steel, ⅛-inch, stiffened by angles and braced light angle beams between its sides and ends. At the top is one filter compartment, so arranged as to cause the discharge from the air pumps to flow over a weir downward through the loofa into a channel, and up another channel and over another weir into a common middle passage to the feed-tank reservoir below. This insures the constant submergence of the filtering material. The capacities of the upper and lower parts are 150 and 350 gallons, respectively. The connections with the first compartment of each filter section are as follows:

Two main air pumps, diameter, inches	9
Filling pipe, diameter, inches	2
Distiller fresh-water connections, diameter, inch	$0\frac{1}{2}$
Vapor pipe, from each filter chamber, diameter, inches	2
Drain pipes, with valves, diameter, inch	$0\frac{1}{2}$

On the feed tank are the following connections:

Overflow, diameter, inches	5
Vapor, diameter, inches	2
Two feed suctions, diameter, inches	9
Drain plug in bottom, diameter, inches	$1\frac{1}{2}$

Water-gauge glasses.
Thermometer.

THE FEED-WATER HEATERS.

There is one feed-water heater in engine room in which the feed water is heated by the exhaust steam. It is of the Normand type and consists of a shell of steel plate to which are bolted two flat tube heads of steel and two spherical water heads of the same metal. The tubes are expanded into the tube sheets. The tubes are of brass, five-eighths of an inch diameter (outside) and No. 18 B.W.G. thick, spaced $\frac{7}{8}$-inch pitch. Inside of each tube is placed a retarder of crimped brass ribbon, $\frac{5}{16}$ inch wide, No. 20 B.W.G. thickness, extending the entire length of the tube. The axis of heater is placed vertically and it is secured to the forward engine-room bulkhead port side. Secured by tap bolts over and to the upper tube plate is a thin, perforated plate of steel serving as a scattering plate and a binder to the ends of the baffles which protrude from the tubes. The holes in plate are $\frac{3}{16}$ inch diameter, pitched $\frac{3}{8}$ inch apart. The holes in the scattering plate are chamfered to prevent sealing the holes by contact with the ends of the baffles. In the lower head is a spiral coil of copper pipe leading from interior of the shell, eighteen inches above the bottom tube head to an exit on the lower head, for the condensation of vapor and the exit of air. There are seven flat baffle plates through which the tubes pass, secured to the inside of the shell by three ears each. Each baffle extends $4\frac{9}{16}$ inches beyond the center line of the heater and

is made of $\frac{1}{8}$-inch steel plate. The feed water enters the lower head, passes through the tubes and leaves through a nozzle on the upper head. The exhaust steam enters the shell near the top of the shell and is drained to a trap. Through the center of the end heads and through a central tube expanded into both tube heads is an axial stay rod with nuts at each end and on the under sides of the end heads. The shell is formed of a single sheet with a longitudinal double-riveted seam, a flange is riveted to each end with two rows of $\frac{5}{8}$-inch rivets. The flange is reduced in thickness within the area of the bolt circle so as to form an expansion joint between the shell and the tube plate. A $\frac{1}{2}$-inch drain hole is provided in the lower head. A gauge glass with fittings attached to shell seven, and twenty inches above lower flange of shell, is installed. The heater is fitted with a safety valve, a pressure gauge and thermometer.

PARTICULARS OF THE HEATER.

Length over all, feet and inches	7–10$\frac{3}{4}$
of shell between tube heads (inside), feet	6–00
Diameter of shell (inside), inches	23$\frac{1}{4}$
Thickness of shell (inside), inch	0$\frac{5}{16}$
Tube sheets, diameter of, inches	30$\frac{1}{4}$
thickness of at tubes, inch	1
flange, inch	0$\frac{13}{16}$
End heads, thickness of throughout, inch	0$\frac{11}{16}$
Central stay, diameter of, inches	1$\frac{7}{8}$
length of, feet and inches	7–10$\frac{3}{4}$
Baffles, shell space, number of	7
thickness of, inch	0$\frac{1}{8}$
Tubes, outside diameter of, inch	0$\frac{3}{8}$
length of, feet and inches	6–2$\frac{1}{4}$
number of	510
material of; composition: admiralty mixture, 10 copper, 29 zinc, 1 tin.	
Nozzle for feed water, diameter, inches	5
Exhaust inlet, diameter of, inches	7
drain to heater, *via* trap, diameter of, inches	2
Spiral air coil, diameter of pipe (copper), inside, inch	0$\frac{1}{2}$
thickness of	No. 14 B.W.G.
length of, feet (about)	74
(Note: A valve bolted to the shell is the upper terminus of this pipe.)	
Heating surface, outside of tubes, square feet	500.69
Area through tubes, retarders fitted, square inches	61 4
Ratio of area, through tubes to outlet	3.12 to 1

Auxiliary Condenser.—This condenser is located in after end of engine room and is fitted with a combined air and circulating pump of Blake type. The cooling surface is 251.5 square feet; number of tubes 280, $\frac{5}{8}$-inch diameter; 18 B.W. G., and 5 feet 8¼ inches long. The condenser has an exhaust connection from the auxiliary exhaust pipe controlled by spring relief valve, and the connection from the exhaust of dynamo turbines is so arranged that it may exhaust direct to auxiliary condenser, while remainder of auxiliary exhaust is turned on feed-water heater or turbines.

AUXILIARY PUMPS.

The engine-room fire and bilge pump is situated on port side immediately abaft the main feed pumps. It is arranged to pump from the bilge and the sea, and discharges to the fire main (flushing system) overboard and, by means of hose connections, to the distillers.

Oil Cooler Circulating Pump.—This is located on starboard side immediately forward of main circulating pump. It has a 3½-inch sea suction, and discharges to oil cooler and to distillers.

Lubricating Oil Pumps.—There are two 6×7×8 oil pumps located on starboard side near oil cooler. They have a 3½-inch suction from oil-drain tank and a 3-inch discharge to oil service through oil cooler.

Drainage Pump.—This is a 3½ × 4 × 4 pump located in compartment A–16. It has two 2-inch suction and 1½-inch hose suction, and a 1½-inch overboard discharge and hose connection. By means of the permanent suction and the hose suction the forward compartment can be pumped out.

OTHER AUXILIARIES.

Air Compressor.—The air compressor, which is built by Platt Iron Works, has a capacity of 20 cubic feet per hour, at 2,500 pounds' pressure and 363 revolutions per minute. The steam cylinder is 9 inches diameter. The air is compressed

in four stages, the diameter of different stages being 8, $4\frac{3}{4}$, $2\frac{3}{4}$ and $1\frac{7}{8}$ inches. The machine is situated in the starboard after corner of engine room. It supplies air at 2,500 pounds' pressure to the air tanks for charging torpedoes, and, through a reducing valve, supplies air to a low-pressure system at 100 pounds' pressure for blowing soot off boiler tubes and for operating pneumatic tools.

Steering Engine.—This is a double-cylinder, $5\frac{1}{2} \times 5\frac{1}{2}$, horizontal engine, built by Hyde Windlass Co. It is located in forward part of pilot house.

EVAPORATING PLANT.

The evaporating plant consists of two vertical evaporators and two vertical distillers. The plant is situated in the starboard forward corner of engine room. The required evaporating capacity when turned on auxiliary exhaust is 3,000 gallons per day. The distilling capacity is 2,000 gallons per day.

Circulating water for the distillers is supplied by the oil-cooler circulating pump, from the fire main or by means of hose connection from the engine-room fire and bilge pump.

There is a vertical evaporator feed pump and a fresh-water pump. A small testing tank is fitted near distillers from which the fresh-water pump takes its suction.

The evaporators have one composition steam head each, to which the tube sheet is bolted. The tubes have return bends in horizontal plane and the divisions in steam-head are arranged so that the steam passes four times through the tubes. The double baffle plate is fitted above the tubes, and besides this, there is a dry pipe. The feed connection has a perforated internal feed pipe and this is arranged to distribute the discharge over a wide area. The steam head has a steam gauge, safety valve and drain connection. The shell has a steam gauge, safety valve, test cocks and gauge glass. The shell is made of galvanized sheets and the clothing of magnesia is lagged with galvanized iron.

PARTICULARS OF EVAPORATOR.

Height of shell over all, feet and inches	5-9
Diameter of shell, thickness, inch	$0\frac{3}{16}$
Thickness of bumped heads, inch	$0\frac{1}{4}$
Heating surface, square feet	75
Number of tubes, brass, 1¼-inch diameter, 13 B.W.G.	
Thickness of tube sheet, inch	$0\frac{3}{4}$
of steam head, inch	$0\frac{5}{8}$
Diameter of steam connection, inches	$1\frac{1}{2}$
vapor connection, inches	$2\frac{5}{8}$
feed connection, inch	$0\frac{3}{4}$
blow pipe, inch	1
drain of steam head, inch	1

Distillers.—The two distillers are located on starboard side abaft evaporators and above the evaporator pumps. The shells are made of a brass tube with flanged ends. The tube sheets are of brass and the bumped ends of copper. The distillers are located vertically, and the drains are arranged so that distilled water can be sent either to ship's tanks or to reserve-feed tanks.

On the upper head is a nozzle for the vapor pipe from the evaporators. The vapor passes through the tubes from top to bottom, while the circulating water zigzags around them from bottom to top, directed by four baffles, through which the tubes pass and which are bolted by ears to the sides of the shell. The circulating-water nozzles are riveted and sweated to the shell. The tubes are expanded and sweated into the tube sheets. The spherical heads are brazed to composition flanges, the tube sheets are thinned to a $\frac{5}{16}$-inch thickness at the flanges, and the shell flanges are secured to the others each by four collar studs, with nuts at both ends and twenty-eight through bolts, all of class B steel. The collar bolts permit the spherical heads to be removed without disturbing tube heads, the through bolts dropping out on removal of the nuts from the ends, while the collar bolts remain to hold the tube sheets.

DIMENSIONS OF DISTILLER.

Cooling surface, square feet	16.5
Shell, brass, inside diameter, inches	7½
thickness	12 B.W.G.
Outside heads, copper, radius of, inches	$4\frac{1}{4}$

Thickness of tube sheets	14 B.W.G.
Diameter of tubes, tinned inside and outside, 18 B.W.G., O. D., inch	$0\frac{3}{4}$
Number of tubes	61
Length of tubes over all, inches	$21\frac{3}{8}$
Outside diameter of flanges and tube sheets, inches	$10\frac{1}{4}$
Vapor connection, diameter, inches	$1\frac{1}{2}$
Drain connection, diameter, inch	1
Circulating-water connection, diameter, inches	$1\frac{1}{2}$

PIPING.

The main steam pipe is of seamless-steel tubing. The branches from the two forward boilers unite to form the starboard steam pipe, and the branches from the two after boilers unite to form the port steam pipe. The branches from boilers are $6\frac{1}{2}$ inches in diameter; the mains are 9 inches in diameter. Each steam pipe has an expansion joint in fireroom and a cut-out valve at forward engine-room bulkhead. Drains are fitted to steam pipe in engine room.

The auxiliary steam pipe is of seamless-steel tubing and takes its supply from the auxiliary stop valve on each boiler steam dome. The branches from boilers are 3 inches diameter. In forward fireroom the pipe is 2, $2\frac{1}{2}$ and 3 inches in diameter; in after fireroom $2\frac{1}{2}$, 3 and $3\frac{1}{2}$ inches in diameter. The pipe leads along starboard side of boiler room and enters engine room on starboard side. In engine room the auxiliary steam pipe has branches leading to the various auxiliaries. Cut-out valves are fitted so that certain parts of piping can be cut out for overhauling.

The auxiliary exhaust pipe is of copper and leads on port side of boiler rooms. In the boiler rooms it takes the exhaust of the anchor and steering engine, drainage pump, the fireroom pumps and the blower turbines. At forward fireroom the pipe is $2\frac{1}{2}$ and 3 inches in diameter, increasing to 7 inches where it takes the exhaust of forward blower, and to 8 inches after taking exhaust of after blower. In engine room the exhaust pipe forms a loop for the various auxiliaries and has connections for discharging to feed-water heater, 7-inch; to auxiliary condenser, 6-inch; to main condensers, 6-inch; to L.P. turbine, via maneuvering steam line to port L.P., 6-inch;

to main H.P. turbine, 6-inch, and to third expansion of each L.P. turbine, 3-inch.

The auxiliary exhaust also has a 2½-inch connection to evaporator vapor pipe.

The exhaust connections to port main condenser and auxiliary condenser and turbo-generators are arranged so that the turbo-generators can have a direct exhaust to auxiliary condenser or to port main condenser, while the rest of auxiliary exhaust is turned on feed heater or turbines. This is very desirable, since it allows the dynamo turbines to operate under a good vacuum both at sea and in port.

Drainage.—The fire and bilge pump in each boiler room and in engine room each have a suction to the bilges of their own compartment. The main circulating pumps each have a 7-inch bilge suction. The forward compartments are pumped by means of the drainage pump in compartment A–17. The after compartments are pumped by hand pump only.

Fire Main.—Each of the three fire and bilge pumps have a 2½-inch discharge to fire main. The fire main is a 2½-inch lead-lined steel pipe and has six 1½-inch hose connections on deck.

The flushing system is supplied from the fire main forward and aft in the wash rooms. Pressure on pipe is regulated by means of regulating valves which relieve when pressure goes above a certain point.

Feed Suction Piping.—A combined suction pipe of copper, 9 inches diameter, leads from the bottom of feed tank to the main feed pumps. Here there are two branches, 4½ inches diameter, leading to the two main feed pumps; a third branch, 6 inches in diameter, leads forward to the boiler rooms, and has a 4½-inch suction to each of the auxiliary feed pumps. There is also a 3½-inch suction to reserve feed tank.

The 9-inch suction pipe also has a 4½-inch branch leading to the port air-pump channelways, by means of which water can be pumped out of condensers with the feed pumps.

Feed Discharge Piping.—The main feed pumps each have a 4-inch discharge which combines into a 5-inch pipe, which

leads through or by-passes heater and has a 2½-inch branch to each main feed connection on boilers.

The auxiliary feed pumps have 4-inch discharge pipes from which 2½-inch branches are taken to the auxiliary feed connections on boilers in their own compartment.

Bottom-Blow and Pumping-Out Piping.—From each lower drum of boiler a 1¼-inch blow pipe leads to a sea valve in each boiler room. A 1¼-inch surface blow from each steam-drum of boiler in compartment also connects into this bottom-blow system. The system also has a 1½-inch hose connection for pumping out boilers by means of the auxiliary feed pump, a section of flexible metallic hose being used to make the connection.

Reserve Feed Piping.—The reserve feed tank, situated in forward end of engine room, has a 3½-inch connection to the combined feed suction just forward of the suction connections for main feed pumps. There is a cut-out valve in feed suction line just abaft the reserve feed-tank connection. Through this connection water from feed tank or from boilers can be run back to reserve feed tank, and any of the feed pumps can pump from the reserve feed tank; the auxiliary feed pumps being able to pump from this tank while the main feed pumps are pumping from the main feed tank.

There is also a ¾-inch make-up feed connection from reserve feed tank to the port condenser.

PROPELLERS.

These are solid manganese-bronze, true screws of uniform pitch. The starboard and center screws are right-handed and the port screw left-handed. The propellers were cast at the foundry of The Hyde Windlass Co., and were machined and polished at builders' works.

Diameter, inches	63
Pitch, inches	58
Disc area, each screw, inches	3,117.2
Helical area, each screw, inches	2,128.8
Projected area, each screw, inches	1,854.6
P÷D	.921

CURTIS STEAM TURBINE (DYNAMO SET).

This small turbine is shown on page 823, Vol. XXI, No. 3, of the JOURNAL, published in August, 1909. The rated capacity of the dynamo is 5 kw.; voltage, 125.

PUMP AND PUMP CONNECTIONS.

No.	Pumps.	Size, Inches.	Kind.	Inches.	Suction pipes from—	Inches.	Discharge pipes to—	Location.
2	Main air.....	12×26×18	Vert., independent.	10	Condenser.....	9	Feed tanks.	Aft. end of eng.room.
2	Main circulating.	7×5 28 in. runner.	Vert., single, centrifugal.	17½ 7	Sea..... Bilge.....	17½ ...	Condenser. Oil cooler, starbd.	Two in eng. room.
1	Aux. condenser.	6×8×8×7	Hor. combined.	4 4	Sea..... Condenser.....	3 4	Feed tank. Overboard.	One in eng. room.
2	Main feed	13×9×16	Vert., piston, simplex.	4½ 3½ 4½	Feed suct. pipe. Reserve feed tanks. Channelways.	4	Main feed disch'ge.	One in each engine room.
2	Aux. feed.....	13×9×16	Vert., piston, simplex.	1½ 4 3½	Bottom blows, hose con'tion. Feed suct. pipe. Reserve feed tanks.	3½ 2	Aux. feed. Overboard.	One in each boiler r'm.
2	Fireroom fire and bilge.	8×5×12	Vert., piston, simplex.	3½ 3½ 1½	Fireroom floors.... Sea ..... Hose connection.	2½ 3 1½	Fire main. Overboard. Hose connection.	One in each boiler r'm.
1	Engine-room fire and bilge.	10×7×8	Vert., piston, simplex.	4 4 1½	Sea ..... Bilge, in engine room. Hose connection.	2½ 3½ 1½ 1½	Fire main. Overboard. Distiller Hose connection. Oil cooler.	One in eng. room.
1	Evap. feed....	3½×4×4	Vert., piston, single.	1½	Sea ..... Dist. circulating water disch.	1	Evap'tors.	Near evaporators.
1	Distiller fresh water.	3½×4×4	Vert., piston, single.	1½	Distilled water tank.	1 1	Ship's ta'ks Reserve feed tanks.	Near evaporators.
2	Oil.....	6×7×8	Vert., piston, simplex.	3½	Bilge oil drain tank.	3	Oil-press. system.	Near oil cooler.
1	Oil cooler, circulating.	6×7×8	Vert., piston, simplex.	3½	Sea	3 1½	Oil cooler. Distillers.	Near oil cooler.
4	Fuel-oil service.	4½×3×4	Vert., piston, duplex.	2	Settling tanks.....	1½	Burners.	Two in each boiler compa't.
2	Fuel-oil tank supply.	6×7×12	Vert., piston, simplex.	4½	Storage tanks..... Overboard.	4 4 4	Set. tanks. Stor. tanks. Overboard.	One in each boiler compa't.
1	For'd bilge...	3½×4×4	Vert., piston, simplex	2	Forward bilges ... Hose connection.	1½	Overboard. Hose connection.	Comp't. A. 26.

TORSION METERS.

Shaft horsepower was calculated from the torsional deflection of the shafts, which was observed through use of the Hopkinson-Thring torsion meter, one of these meters being applied to each turbine shaft. A description of this instrument, with illustrations showing essential parts, will be found at the end of this article.

SHAFT CALIBRATION DATA (PAULDING.)

	Shaft 1.	Shaft 2.	Shaft 3.
Length of shaft calibrated, feet and inches	10–05	25–02⅝	10–01⅝
Diameter of shaft, outside, inches	00–06⅝	00–06⅝	00–06⅝
inside, inches	00–03½	00–03½	00–03½
Foot-pounds for one inch torque on 30-inch radius	48,766.8	24,059.7	50,579.4
Length of shaft which torsionmeter records, inches	25.5	25.5	25.5
Constant (K) of torsionmeter	0.0713	0.07002	0.07009

S.H.P. $= K \times$ r.p.m. $\times$ reading of torsion meter.

WATER CONSUMPTION OF AUXILIARIES.

On the *Drayton's* trip to Boston Navy Yard for docking, some water-consumption tests of auxiliaries were made. While main engines were shut off the auxiliaries were operated at the number of revolutions necessary under conditions for obtaining full speed, 25, 16 and 14 knots, and the water consumption measured in each case. From these results a curve of water consumption of auxiliaries was plotted, as shown on Plate VIII.

The water consumption of the blowers was obtained on the *Paulding*. First auxiliaries, including both blower turbines, were operated, the after blower at 1,850 revolutions; air pressure, 3¼ inches, and water measured. Then the after blower turbine was shut down and the water again measured. The difference between these two results represents the water consumption of one of the blower units. This was found to be 7,758 pounds of water per hour.

The horsepower of the blower unit at full speed, 1,850 revolutions, is about 170, and this result would indicate that these blower turbines require 45 pounds of water per H.P. per hour, with back pressure at 20 pounds absolute.

Another test, made in similar manner, but with blower operating at 1,500 to 1,600 revolutions per minute, and air pressure at $1\frac{1}{8}$ inches, gave a water consumption of 5,760 pounds per hour. The water consumption per horsepower at this speed would be about 42 pounds. The back pressure

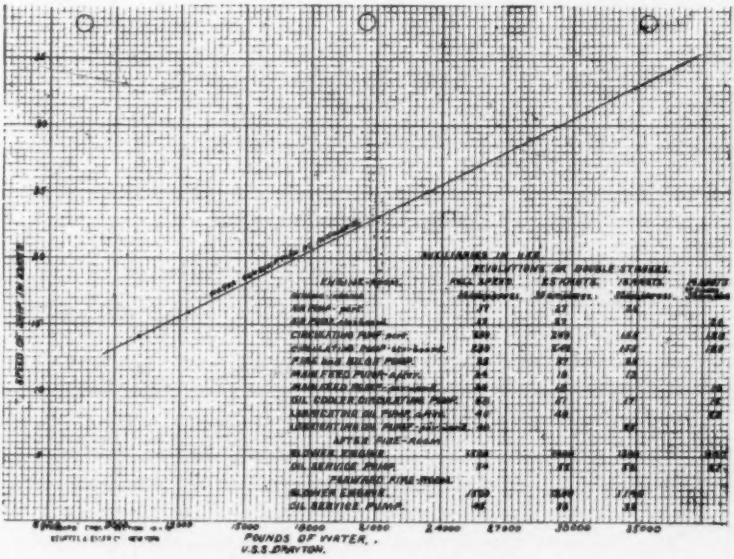


Plate VIII.—U. S. S. DRAYTON.

STEAM CONSUMPTION OF AUXILIARY MACHINERY AT VARIOUS SPEEDS.

was also about 20 pounds absolute. The horsepower assumed above is, however, only approximate. The indications are that though the water rate per horsepower is approximately the same or only slightly higher than with reciprocating engines, the power necessary to supply the air is considerably

greater than it would be with reciprocating engines operating at about 800 revolutions per minute.

REVERSING AND BACKING TESTS ("DRAYTON").

With helm amidships and engines making approximately 29½ knots, the engines were reversed. Time from signal to dead in water, 57.5 seconds.

Time taken in engine room :	*Starboard.*	*Port.*
Began turning over astern, seconds	11.0	11.8
Full speed astern, seconds	41.5	41.5
Revolutions per minute	400.0	400.0

The signal for full speed ahead was then made.

	Starboard.	*Port.*
Began turning over ahead, seconds	8 4	7.8
Full speed ahead, seconds	49.5	49.5

STANDARDIZATION TRIALS.

Paulding.—The standardization runs were made on August 23, 1910, on the Rockland, Me., course. The weather was fair and pleasant, with light to gentle breezes from S.S.E. The sea was smooth, with short swell from southward. Twenty-six runs were made. On the standardization runs the same combination of turbines, for corresponding speeds, as would be used on the 4-hour full-speed, the 25-knot and 16-knot trial. Runs 1, 2 and 3 (speed, 11.088 knots) were made with only the exhaust from auxiliaries entering turbines.

The curve of speed, revolutions and horsepower, deduced from the data on standardization runs, is given on Plate IX.

Average displacement on five high runs, 716.5 tons.

Drayton.—The standardization runs were made on October 11, 1910, on the Rockland, Me., course. The weather was fair, with stiff breezes from W.S.W. to N.W. There was a moderate, choppy sea.

Owing to wind and choppy sea the conditions were not very favorable for obtaining high speed on standardization runs.

TURBINE DATA (ONE SHEET FOR ENGINE ROOMS) Standardization TRIALS.

U. S. S. "Paulding", Course off Rockland Me., on Aug. 23, 1910

NO. OF RUN.	TIME. H.	M.	S.	REVOLUTIONS PER MINUTE. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	AVERAGE ALL SHAFTS.	STEAM PRESSURES. ENGINE ROOM (GAGE). MAIN STEAM.	AUXILIARY STEAM.	AUXILIARY EXHAUST.	IN STEAM BELT (ABSOLUTE). MAIN H. P. TURBINE.	CRUISING TURBINES. H.P.	L.P.	L. P. TURBINES. P.	S.	OIL SANDS GAGE.	PRESSURE IN OILING SYSTEM (GAGE).	VACUUM IN INCHES OF MERCURY. IN ENGINE ROOM. P.	S.	TEMPERATURES. INJECTION. P.	S.	DISCHARGE. P.	S.	AIR-PUMP DISCHARGE. P.	S.	HEATERS (FEED). FEED TO. P.	S.	FEED FROM. P.	S.	OUTSIDE AIR.	TORSION METER READINGS. K= SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	BRAKE HORSEPOWER. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	TOTAL.
1				268.76	224.28	258.44	247.98	245		5			30	6.2	5.0	1	2	29.0	29.0	100	59		70	72	82	92			232		15	7.0	8.0	281	110	144	535
2				272.57	232.22	264.57	256.29	250		6			29	6.2	5.5	1	2	29.0	29.0	101	59		67	68	80	91			226		15	7.5	8.5	292	122	158	572
3				277.05	231.30	259.40	259.40	248		6			29	6.2	5.5	1	2	29.9	29.9	101	59		68	66	74	82			230		16	7.5	11.0	316	123	218	657
4				418.02	375.39	394.76	396.37	245		5	29	110	60	9.0	8.5	1	2.5	29.9	29.9	101	59		69	67	70	86			208		30	9.5	22.0	883	210	609	1702
5				408.19	370.39	393.21	390.93	245		9	29	110	60	9.0	8.5	1	2.	29.9	29.9	101	59		68	67	70	79			214		30	9.5	22.0	875	206	606	1687
6				412.98	374.01	396.89	394.96	241		9	29	110	60	9.0	8.5	1	2.	29.9	29.9	101	59		68	69	70	78			222		30	9.0	22.0	881	198	610	1689
7				406.32	369.97	396.90	391.06	235		2	45	215	110	12.5	11.0	1	2	29.8	29.8	100	59		72	72	73	78			218		42	12.5	36.0	1016	582	1219	3237
8				513.26	466.57	502.87	494.57	247		4	45	215	108	12.5	11.5	1	2	29.7	29.8	100	59		71	72	72	80			218		39.5	12.5	36.0	1445	509	1271	3225
9				506.18	461.06	502.81	490.68	250		4	45	215	102	12.7	11.2	1	2	29.7	29.8	99	59		71	74	72	77			218		39.0	12.5	36.0	1419	584	1269	3272
10				614.88	558.60	598.99	590.82	247	Same as main steam.	4	81		205	20.0	17.5	1	4	29.5	29.9	99	59		74	74	74	75			210		59.5	26.0	38.5	2609	1098	1481	5285
11				611.31	556.01	596.98	588.10	244		4	81		205	19.5	17.5	1	4.5	29.3	29.3	99	59		75	72	76	77			208		60.0	26.5	38.5	2617	1098	1476	5401
12				618.01	560.72	602.92	593.88	245		4	81		205	18.5	17.5	1	4.5	29.3	29.3	99	58		76	75	76	76			212		58.5	26.5	39.5	2578	1119	1621	5428
13				661.87	594.39	636.72	607.66	250		98	100		255	22.5	21.0	1	4.0	29.2	29.2	99	58		74	76	76	77			210		67.0	44.0	44.5	3162	1770	1830	6762
14				666.77	599.06	636.36	611.71	250		4	100		255	22.5	21.0	1	4.0	29.2	29.2	99	58		74	76	76	76			208		66.5	44.0	46.0	3157	1784	1862	6803
15				662.98	596.85	638.11	609.65	250		4	100		255	22.5	21.0	1	4.0	29.3	29.3	99	58		74	74	76	78			208		66.5	44.0	46.0	3148	1774	1858	6780
16				727.58	706.25	715.79	716.54	245		3	180			38.0	36.5	1.5	5.0	29.0	28.5	99	58		82	82	86	86			210		67.0	73.0	74.0	3476	3610	3713	10799
17				702.61	695.37	701.51	701.50	248		2	180			38.0	38.0	1.5	4.0	29.0	29.0	99	57		81	80	85	86			194		64.0	68.0	71.0	3229	3311	3491	10031
18				717.59	700.96	709.29	709.28	245		3	175			38.0	37.5	1.5	4.0	29.0	28.5	99	57		82	85	84	88			208		63.0	68.0	72.0	3224	3337	3580	10140
19				769.76	740.84	769.85	756.65	243		2	205			43.0	41.5	1.5	4.0	29.0	28.5	99	57		83	83	89	89			194		73.0	76.0	78.5	4007	3942	4177	12126
20				766.58	745.99	769.52	760.70	245		4	205			42.0	42.5	1.5	5.0	29.0	28.5	99	57		83	85	89	89			200		73.0	76.0	78.5	3990	3916	4152	12058
21				771.26	741.74	760.91	757.97	240		3.5	205			42.0	42.5	1.5	4.0	29.0	28.5	99	57		83	90	89	92			200		72.5	76.5	78.5	3987	3923	4187	12147
22				926.41	864.91	930.88	910.73	245		3.0	265			60.0	60.0	2.5	4.0	28.0	27.5	99	57		91	93	96	96			193		84.0	89.0	100	5569	5390	6425	17524
23				931.35	853.81	924.65	903.27	240		1	255			60.0	58.0	2.5	4.5	30.0	27.5	100	56		94	93	96	95			188		85.0	88.5	101	5647	5291	6046	17482
24				944.59	866.36	937.58	916.18	240		4	255			60.0	60.0	2.5	4.0	29.5	27.0	99	56		94	94	96	96			188		85.0	89.0	102	5745	5399	6203	17827
25				947.92	866.99	941.17	919.69	250		4	255			62.0	60.5	2.5	6.0	29.5	27.0	102	56		94	93	96	96			186		86.0	89.5	102	5813	5426	6259	17998
		.26		954.60	877.85	926.76	919.76	250		3	255			62.5	61.5	1.5	7.0	29.5	27.0	99	56		90	90	90	90			184		90.	90.	102	6127	5532	6626	18285

Running Idle

TURBINE DATA (ONE SHEET FOR ENGINE ROOMS) STANDARDIZATION TRIALS

U. S. S. DRAYTON, Course off ROCKLAND, MAINE., on OCT. 11, 1910.

NO. OF RUN.	TIME. H.	M.	S.	REVOLUTIONS PER MINUTE. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	AVERAGE ALL SHAFTS.	STEAM PRESSURES. ENGINE ROOM (GAGE). MAIN STEAM.	AUXILIARY STEAM.	AUXILIARY EXHAUST.	IN STEAM BELT (ABSOLUTE). MAIN H. P. TURBINE.	CRUISING TURBINES. H.P.	I.P.	L. P. TURBINES. P.	S.	ON GLANDS GAGE.	PRESSURE IN OILING SYSTEM GAGE.	VACUUM IN INCHES OF MERCURY. P.	S.	TEMPERATURES. IN ENGINE ROOM.	INJECTION. (WATER FROM OIL COOLER) P.	S.	DISCHARGE. P.	S.	AIR-PUMP DISCHARGE. P.	S.	HEATERS (FEED). OIL COOLER. OIL TO P.	OIL FROM S.	FEED FROM: P.	S.	OUTSIDE AIR.	TORSION METER READINGS. K=.00117. SHAFT NO. 1.	K=.00116. SHAFT NO. 2.	K=.00111. SHAFT NO. 3.	BRAKE HORSEPOWER. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	TOTAL.
1				264.21	248.36	277.36	263.65	262	245	6	103.35	26		6	6	1.5	4	29.13	29.80	54	55	64	74	74	74	70	67	230	46	5.5	11	10	105	187	194	486
2				263.11	248.07	261.41	255.86	260	245	6	10.529	24		5.5	5.5	1.5	4.5	29.22	29.80	"	55	64	74	76	76	70	67	232	46	5	10	9	85	166	165	476
3				263.66	247.15	272.03	262.95	268	245	6	105	38	30	6	6	1.7	4.5	29.23	29.80	"	55	66	73	73	74	70	67	236	45	5.25	11	9.5	109	185	185	479
4				345.78	340.21	346.39	344.62	270	252	6	23	110	55	9	10	1.7	4	29.4	29.80	"	55	66	70	70	74	72	68	236	46	16.5	12	23	467	254	638	1359
5				370.23	341.48	374.21	353.85	262	245	7	25	110	55	9	9.5	1.8	4	29.0	29.81	"	56	66	70	70	74	72	69	236	46	15	13	22	400	276	585	1261
6				281.95	312.01	396.73	336.06	262	245	7	25	110	55	9	9.5	1.7	4.5	28.73	29.80	"	56	66	70	70	74	74	70	238	48	16.5	13	23	466	281	636	1383
7				424.38	386.48	476.70	445.19	265	245	6	37	210	100	12	12.5	1.5	3.7	28.9	29.80	"	56	68	70	70	74	75	70.5	232	54	27	17.5	32.5	922	461	1084	2467
8				428.34	388.24	477.50	448.13	267	245	6	37	210	102	13	13	1.5	4.5	28.9	29.80	"	56	68	70	76	72	77	72	228	52	27.5	18	38.5	957	477	1120	2548
9				411.15	381.86	467.53	487.18	260	243	6	325	210	102	13	13.5	1.5	4.5	28.9	29.82	"	56	74	74	72	72	78	73	232	55	27.5	18	34.5	953	475	1177	2605
10				581.26	587.74	581.95	583.65	267	245	6	72			196	18.5	18.5	1.5	4.5	28.3	29.84	"	56	74	74	72	72	80	76	236	54	46.25	34.5	36	1990	1194	1340	4524
11				582.26	581.47	582.73	582.49	270	242	6	75			192	18.5	18.5	1.5	4.5	28.5	29.84	"	56	74	74	76	76	82	76	234	56	49	34.5	38	2093	1179	1425	4697
12				554.38	557.85	549.45	550.56	268	242	6	25			192	18.5	18.5	1.5	6	28.9	29.86	"	56	74	74	72	72	82	78	230	57	48.5	34	38.5	2093	1177	1576	4786
13				644.23	627.61	580.33	617.39	270	255	6	97			262	20.5	20.5	1.5	5	28.5	29.86	"	56	74	74	72	72	86	80	230	58	58.5	44.5	45	2715	1706	1829	6250
14				653.39	656.60	656.41	660.45	270	247	5	100			260	21	21	1.6	5	28.8	29.86	"	56	74	74	72	72	86	80	234	57	58	44.5	46	2771	1735	1888	6394
15				653.39	656.32	656.32	660.67	270	250	6	100			260	21	21	1.6	4.5	28.7	29.86	"	55	74	74	72	72	87	80	234	56	59.25	44	46.5	2778	1689	1914	6401
16				708.50	694.48	721.88	708.27	266	243	6	180			38	22.5	1.6	5	28.2	29.86	"	56	78	78	80	80	90	81	240	56	59	24	26.5	3009	3503	3865	10377
17				706.28	694.83	723.89	708.67	266	245	6	180			38	22.5	1.6	5	28.7	29.87	"	56	78	78	80	80	90	82	240	58	59	24	28	2998	3490	3946	10434
18				710.38	694.43	724.12	709.31	252	232	7	180			38	22.5	1.6	5	28.5	29.88	"	56	80	80	80	80	90	82	248	56	59	24	28	3019	3402	3853	10274
19				748.48	745.01	788.27	760.62	265	225	7	215			44	24	1.8	4.5	28.0	29.86	"	57	84	84	82	82	96	84	240	58	64	27	26.5	3485	4448	4778	12381
20				766.68	748.99	789.20	769.58	260	220	7	245			45	44	1.8	4.5	28.2	29.73	"	56	84	84	80	80	91	82	240	57	64	28	28.5	3534	4485	4787	12816
21				764.71	747.03	791.89	767.90	260	220	7	245			45	40	1.5	4.5	28.2	29.78	"	56	86	86	82	82	92	84	240	60	64	27	26.5	3524	4430	4794	12748
22				942.35	854.76	973.64	923.92	257	220	7	268			63	63	1.5	4.5	28	27.1	86	"	56	92	92	88	84	95	88	240	60	82.5	96.5	102.5	5626	5822	7318	18766
23				944.17	852.80	977.31	924.76	262	225	7	252			65	65	1.5	5.5	28	27	86	"	56	92	92	86	86	94	88	240	60	82.5	97	102.5	5688	5638	7353	18679
24				947.17	851.46	973.44	924.02	262	220	8	253			62	63	1.5	6	28.2	27.9	86	"	56	92	92	86	86	94	89	240	60	83.5	97	102.5	5694	5629	7324	18647
25				913.38	866.72	986.15	922.70	270	230	6	255			65	65	1.5	6	28	27.2	86	"	56	92	92	90	90	92	89	240	60	84	98	102.5	5769	5748	7491	19009
	26			462.47	465.27	497.29	475.53	265	225	6	255			65	65	1.5	6.5	28	27.2	86	"	56	94	94	90	90	94	90	240	59	83.5	98	102.5	5713	5726	7427	18866

Running Idle

Twenty-six runs were made in the same general manner as in the case of the *Paulding*.

The curve of speed, revolutions and horsepower, deduced from the data on standardization runs, is given on Plate X.

FOUR-HOURS' FULL-SPEED TRIAL.

Paulding.—This trial was begun at 9.45 A. M.. September 2, 1910, and ended at 1.45 P. M. same day. The course was off Seguin Island. The weather was fair and pleasant, with gentle to moderate breeze from N.E. There was a light swell from eastward.

Average displacement on trial, 711 tons. Mean draught on trial, 8 feet $1\frac{3}{4}$ inches.

Drayton.—This trial was begun at 2 P. M., October 14, and ended at 6 P. M. the same day. The course was off Seguin Island. The weather was cloudy, overcast and misty, with light breeze from N.E. to North.

There was a long, heavy swell from S.W. at beginning of trial, sea becoming smooth at end of trial.

Average displacement on trial, 721 tons. Mean draught on trial, 8 feet .02$\frac{9}{16}$ inches.

PERFORMANCE.—FOUR-HOURS' FULL-SPEED TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Paulding.	*Drayton.*
Average steam pressures:		
At boilers, pounds, gauge	251.67	254.0
engines, pounds, H.P., gauge	237.5	247.6
M.H.P. turbine, pounds absolute	250.95	250.1
Starboard L.P. turbine, pounds, absolute	59.4	52.7
port L.P. turbine, pounds, absolute	59.89	52.3
On turbine glands, pounds, gauge	1.67	1.5
auxiliary exhaust, pounds, gauge	1.38	6.0
Vacuum in condensers, inches of mercury, starboard	27.6	26.81
port	27.9	28.01
pressure in lubricating oil system, pounds, gauge.	4.87	4.8
fuel-oil system, pounds, gauge	130.0	219

Temperatures. (*Average of one-half hourly observations.*)

	Paulding.	*Drayton.*
Injection, degrees F.	56.2	52.0
Discharge, degrees F.	92.0	85.5
Feed tank, degrees F.	96.4	83.0
water, leaving heater, degrees F.	187.2	204.0
Engine room, working level, degrees F.	96.7	86.5
Firerooms, working level, degrees F.	98.3	94.0
Outside air, degrees F.	...	52.0
Smoke stacks, average, degrees F.	...	1,293*
Fuel oil to burners, degrees F.	77.5	60.7
Per cent. moisture, main steam line	4.3	3.0
Oil to oil cooler, degrees F.	...	83
from oil cooler, degrees F.	...	79
Water from oil cooler, degrees F.	...	55

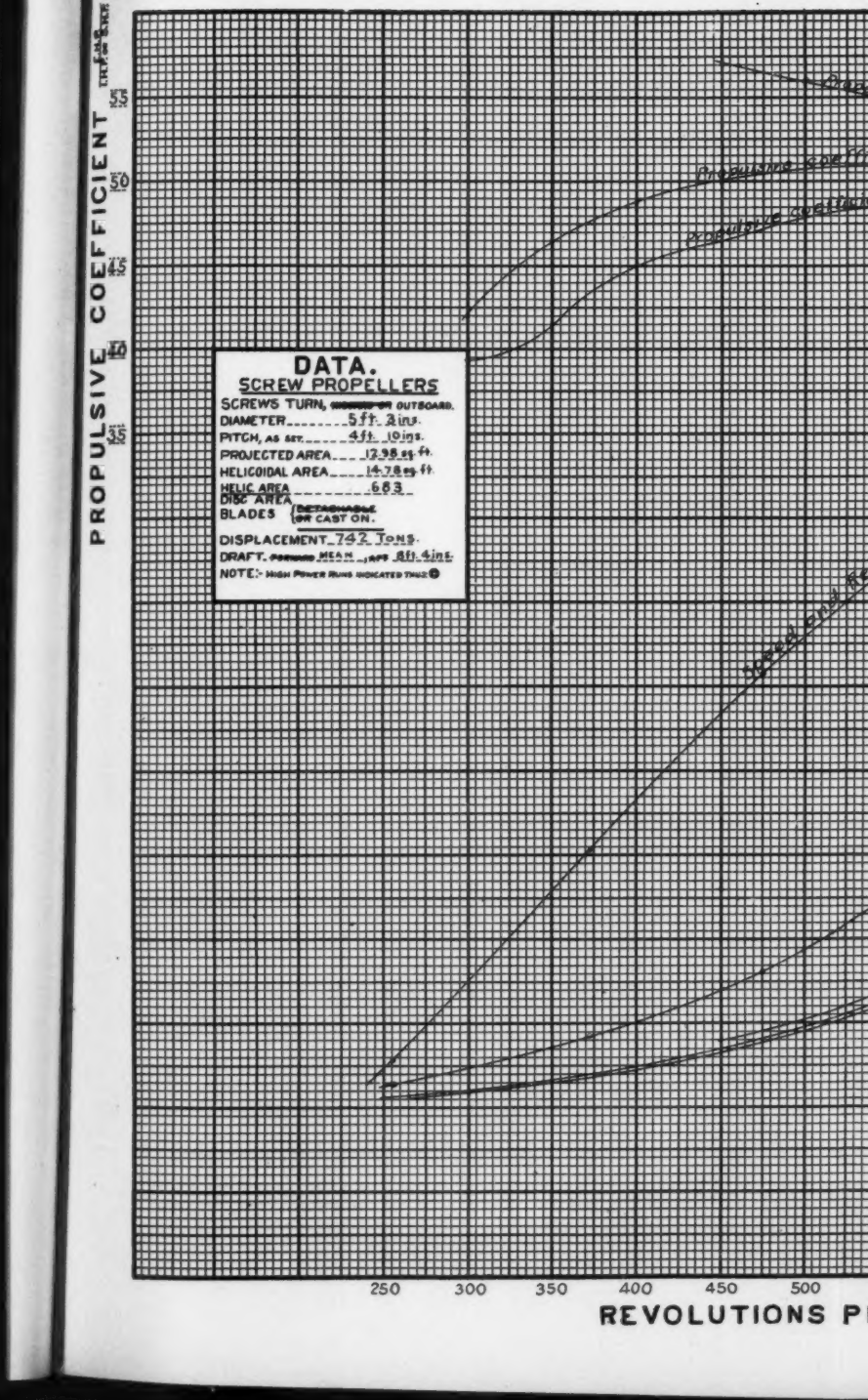
* Pyrometers considerably out on this and other runs.

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

	Paulding.	*Drayton.*
Average revolutions, shaft No. 1	933.559	844.5
No. 2	847.699	812.1
No. 3	927.762	877.1
average	903.007	844.5
Pumps, main air, starboard	31.8	57.0
port	45.8	57.0
circulating, starboard	292.2	303.0
port	272.0	296.0
feed, forward	34.4	28.8
after	34.4	32.0
fire and bilge	28.4	14.0
lubricating, oil pump, No. 1.	58.0	61.0
No. 2	60.1	10.0
dynamo condenser, air and circulating		
Oil cooler, circulating	68.8	60.0
Forced-draft blower engines, forward	1,531.0	1,598.0
after	1,512.0	1,548.0
Speed of ship, in knots per hour	32.801	30.83
Slip of propeller, in per cent. of its own speed, based on Mean pitch :		
Shaft No. 1	26.34	23.5
No. 2	18.877	20.4
No. 3	25.878	26.3
Mean	23.698	23.4
Air pressure in firerooms, in inches of water, mean	4.8	4.8

SHAFT HORSEPOWER.

Shaft No. 1	5,604.4	4,377
No. 2	5,169.4	5,386
No. 3	6,225.1	5,761
Total shaft horsepower from standardization curve	17,393.0	15,524


PROPULSIVE COEFFICIENT
55
50
45
40
35
DATA.
SCREW PROPELLERS
SCREWS TURN, OUTBOARD.
DIAMETER 5 ft. 3 ins.
PITCH, AS SET 4 ft. 10 ins.
PROJECTED AREA 12.98 sq. ft.
HELICOIDAL AREA 14.78 sq. ft.
HELIC AREA / DISC AREA .663
BLADES DETACHABLE / OR CAST ON.
DISPLACEMENT 742 TONS.
DRAFT. MEAN 8 ft. 4 ins.
NOTE:- HIGH POWER RUNS INDICATED THUS ⊕
250
300
350
400
450
500
REVOLUTIONS P

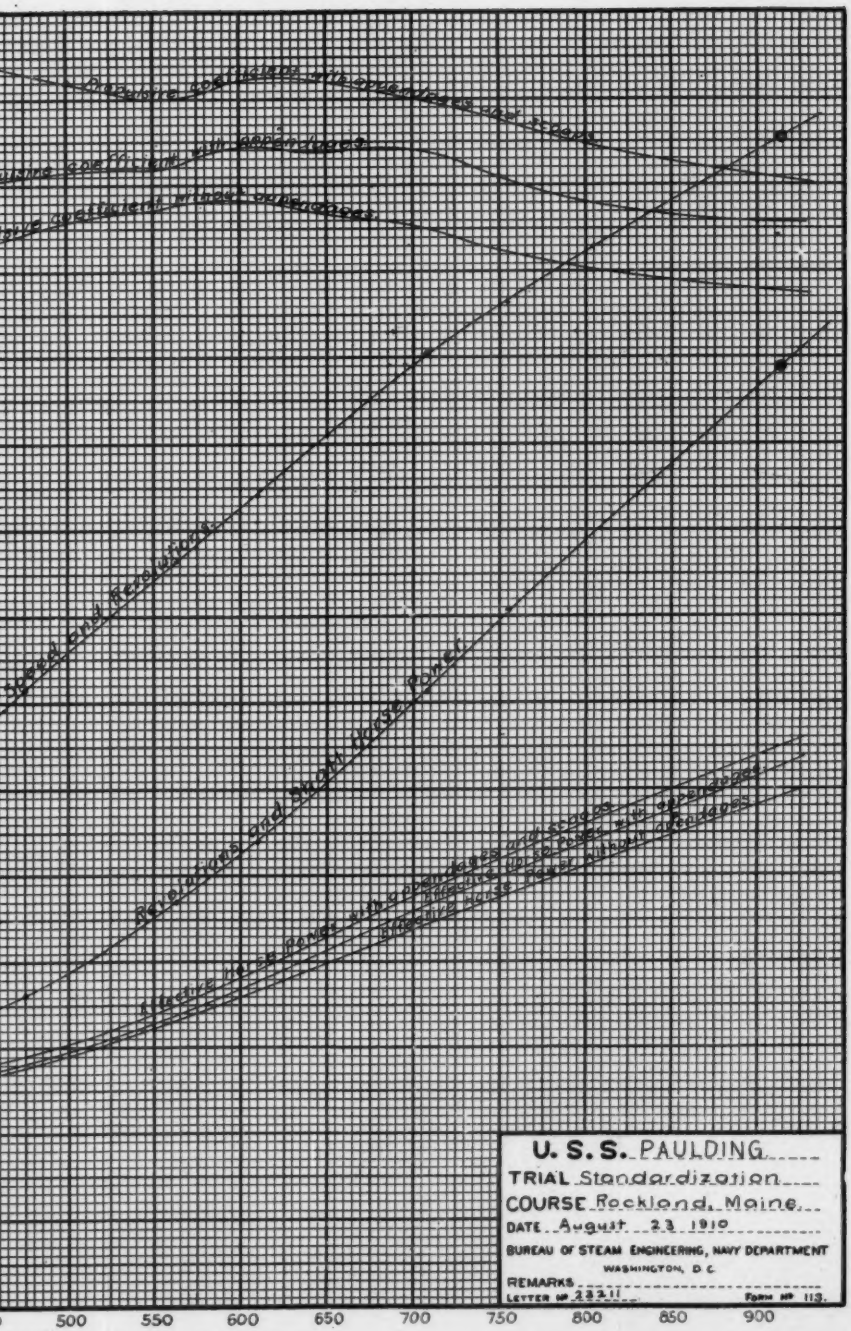
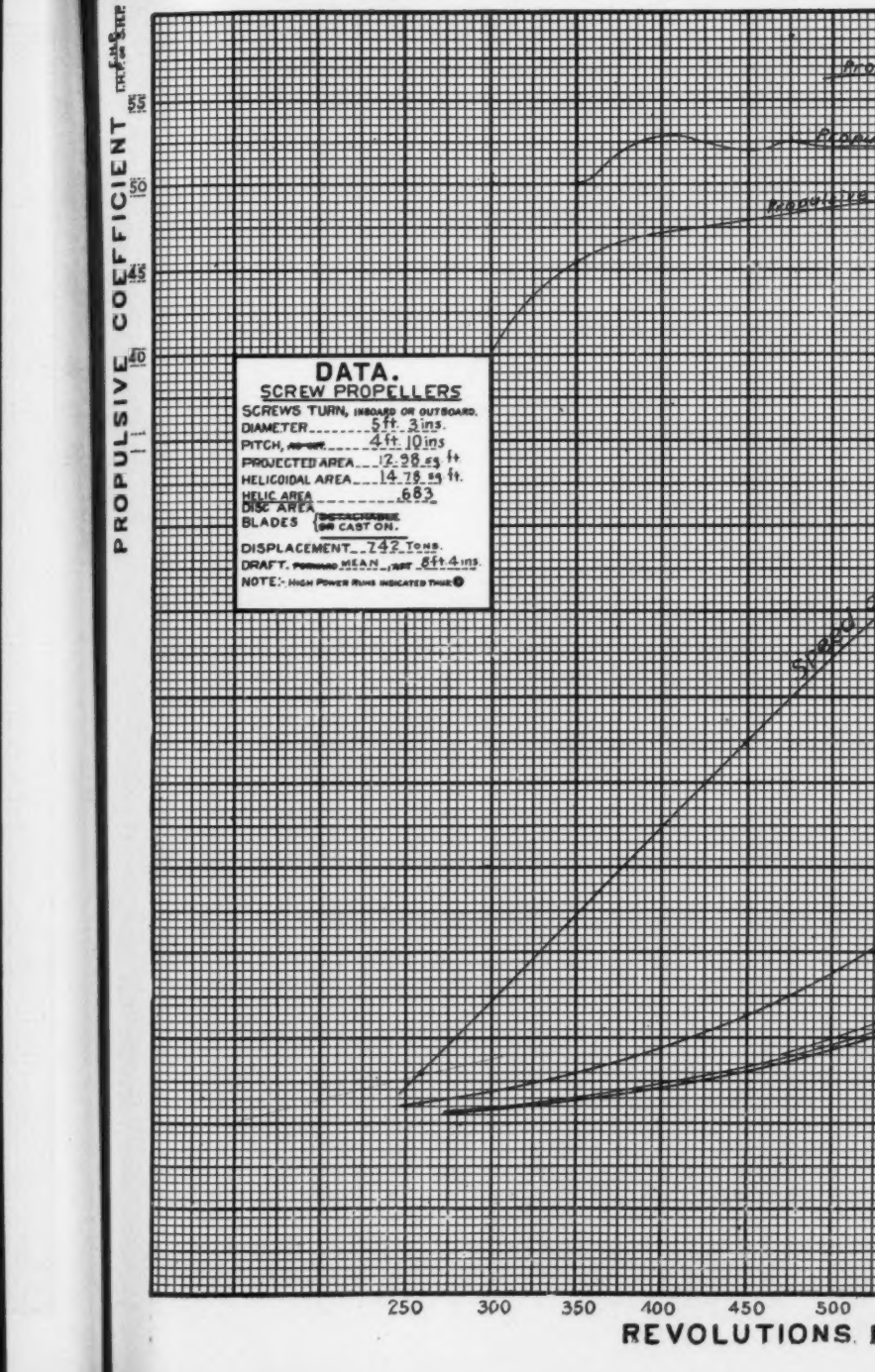

U. S. S. PAULDING
TRIAL Standardization
COURSE Rockland, Maine.
DATE August 23 1910
BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT
WASHINGTON, D.C.
REMARKS
LETTER NO. 23311
Form No. 113.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
0
2000
4000
6000
8000
10000
12000
14000
16000
18000
10
12
14
16
18
20
22
24
26
28
30
32
34
500
550
600
650
700
750
800
850
900
TIONS PER MINUTE.

Plate IX.


PROPULSIVE COEFFICIENT
DATA.
SCREW PROPELLERS
SCREWS TURN, INBOARD OR OUTBOARD.
DIAMETER 5ft. 3ins.
PITCH, AS-OUT 4ft. 10ins
PROJECTED AREA 12.98 sq ft
HELICOIDAL AREA 14.78 sq ft.
HELIC AREA .683
DISC AREA
BLADES {DETACHABLE. OR CAST ON.
DISPLACEMENT 742 Tons.
DRAFT. FORWARD MEAN AFT 8ft 4ins.
NOTE:- HIGH POWER RUNS INDICATED THUS ⊕
250
300
350
400
450
500
REVOLUTIONS.

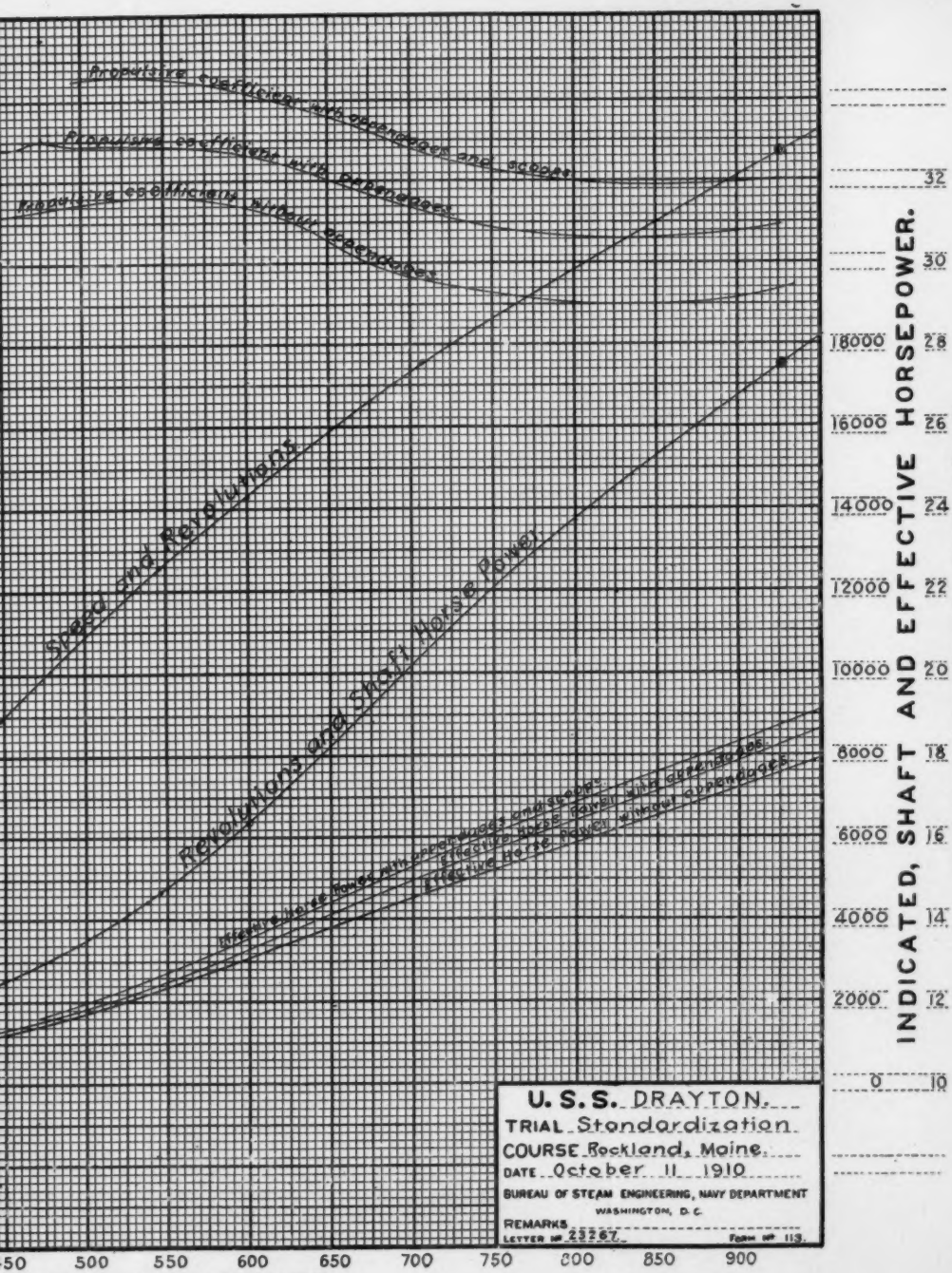

Propulsive coefficient with superheaters and screws
Propulsive coefficient with superheaters
Propulsive coefficient without superheaters
Speed and Revolutions
Revolutions and Shaft Horse Power
Indicated Horse Power with superheaters and screws
Indicated Horse Power with superheaters
Indicated Horse Power without superheaters
Effective Horse Power without superheaters
U. S. S. DRAYTON.
TRIAL Standardization.
COURSE Rockland, Maine.
DATE October 11 1910
BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT
WASHINGTON, D. C.
REMARKS
LETTER NO 23267.
Form No 113.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
0
2000
4000
6000
8000
10000
12000
14000
16000
18000
10
12
14
16
18
20
22
24
26
28
30
32
500
550
600
650
700
750
800
850
900
UTIONS PER MINUTE.

Plate X.

OIL.

B.T.U. per pound of oil	19,659	19,516
Pounds used, per hour	18,367	18,644
Knots run per ton of oil	4	3.704

DEDUCED DATA.

Square feet of heating surface, per S.H.P.	1.11	1.24
Cubic feet of furnace volume, per S.H.P.	.077	.087
Pounds of oil per S.H.P. per hour, main shafts	1.056	1.201
cubic foot of furnace volume, per hour	13.66	13.88
square foot heating surface, per hour	.91	.91
Cooling surface (main condenser), square feet per S.H.P.	.552	.618

WATER.

Pounds per hour, total	218,655	200,899
total per hour per S.H.P. of main engines	12.511	12.941
of water evaporated per pound of oil	11.9	10.77
used by auxiliaries	33,000	(*)

*Approximate. Auxiliary exhaust open to feed heater and condenser.

TWELVE-HOURS' TRIAL AT SIXTEEN KNOTS.

Paulding.—This trial was begun at 5:15 A. M. August 24, 1910, and ended at 5:15 P. M. the same day. The course was along the coast of Maine from Monhegan Island to Seguin Island, to Cape Neddick, to Cape Elizabeth, to Seguin Island, to Monhegan Island, then back to Seguin Island. The weather was generally overcast and cloudy, with haze and fog on horizon; gentle to stiff breeze from S.S.W. There was a short, choppy sea from S.S.W.

Average displacement on trial, 727.5 tons. Mean draught on trial, 8 feet 3⅝ inches.

Drayton.—This trial was begun at 5.05 A. M. October 12, 1910, and ended at 5:05 P. M. the same day. The course was along the coast of Maine from Monhegan Island to Isle of Shoals and back to Seguin Island. The weather was clear, with fresh to strong breezes from N.W. There was a short, choppy sea from N. W.

Average displacement on trial, 727 tons. Mean draught on trial, 8 feet 3 inches.

PERFORMANCE.—TWELVE-HOURS' TRIAL, AT SIXTEEN KNOTS.

Steam Pressures. (*Average of hourly observations.*)

	Paulding.	*Drayton.*
Average steam pressure of boilers, pounds gauge	225.0	...
at engines, pounds gauge	220.0	228.0
H.P. cruising turbine, lbs. abs.	133.0	161.0
I.P. cruising turbine, lbs. abs.	67.0	77.0
H.P. turbine, pounds absolute	31.0	29.0
starboard L.P. turbine, lbs. abs.	7.9	8.4
port L.P. turbine, pounds abs.	7.6	8.4
on turbine glands, pounds gauge	1.0	1.5
auxiliary exhaust, pounds gauge.	4.3	8.0
Vacuum in condensers, inches of mercury, starboard	29.5	28.4
port	29.5	28.6
Pressure in lubricating-oil system, pounds gauge	4.0	4.1
fuel-oil system, pounds gauge	213.3	222

Temperatures. (*Average of hourly observations.*)

Injection, degrees F	59.0	52.0
Discharge, degrees F	...	85.0
Feed tank, degrees F	71.0	83.0
water leaving heater, degrees F	222.0	204.0
Engine room, working level, degrees F	...	86.5
Firerooms, working level, degrees F	...	94.0
Outside air, degrees F	...	52.0
Smoke stacks, average, degrees F	...	828
Fuel oil to burners, degrees F	68.7	60.7
Oil to oil cooler, degrees F	88.0	83.0
from oil cooler, degrees F	81.0	79.0
Water from cooler, degrees F	65.0	55.0
Per cent. moisture, main steam line	3.5	2.0

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, shaft No. 1	390.5	393.9
2	330.6	344.4
3	384.1	393.4
Mean revolutions, both engines, per minute	368.4	377.4
Pumps, main air	25.0	27.4
circulating, starboard	153.0	171.0
port	156.0	162.0
feed, forward	13.0	12.0
after	...	...
fire and bilge	25.0	27.0

	Paulding.	*Drayton.*
Pumps, lubricating-oil, No. 1	...	25.0
2	24.0	...
auxiliary condenser, air and circulating	...	...
oil-cooler, circulating	17.0	31.2
Blower engines forward	1,242.0	...
after	...	1,074.0
Speed of ship, in knots per hour	15.858	16.08
Slip of propeller, in per cent. of its own speed, based on mean pitch :		
Shaft No. 1	14.8	14.4
No. 2	0.58	2.1
No. 3	13.4	14.3
Mean	9.59	10.3
Air pressure in firerooms, in inches of water, mean	4.2	3.1

SHAFT HORSEPOWER.

Shaft No. 1	682.0	421.0
No. 2	181.0	419.0
No. 3	579.0	585.0
Total S.H.P. from standardization curve	1,442.0	1,425.0

OIL.

Kind and quality used on trial :		
Paraffin base, specific gravity	0.8759	0.8789
flash point, degrees	252.0	260.0
B.T.U. per pound of oil	19,659.0	19,516.0
Pounds, per hour	2,853.5	2,554.7
Knots run per ton of oil	12.5	14.1

DEDUCED DATA.

Square feet of heating surface per S.H.P.	3.349	3.22
Pounds of oil per S.H.P. per hour	1.979	1.701
Cooling surface (main condenser), square feet per S.H.P.	6.66	6 39

WATER.

Pounds per hour, total	30,967.0	32,900.0
(total) per hour per S.H.P. of main engines	21.475	21.91
of water evaporated per pound of oil	10.85	12.88
used by auxiliaries	12,500.00	*12,500.00

* Approximate. Auxiliary exhaust open to M.H.P. turbine.

TWELVE-HOURS' TRIAL AT TWENTY-FIVE KNOTS.

Paulding.—This trial was begun at 6.15 A. M. August 25, 1910, and ended at 6.15 P. M. the same day. The course was from Seguin Island to Cape Elizabeth, to Thatchers Island,

thence to Cape Elizabeth and then to Seguin Island, and the run repeated. The weather was hazy, with moderate breeze from S.W. There was a light swell from southward.

Average displacement, on trial, 712 tons. Mean draught on trial, 8 feet 2 inches.

Drayton.—This trial was begun at 5.50 A. M. October 13, 1910, and ended at 5.50 P. M. the same day. The course was from Seguin Island to Thatchers Island, thence to Bar Harbor, and then back to Seguin Island. The weather was clear and pleasant, with light to moderate breezes from N.N.W. Sea smooth.

Average displacement, on trial, 720 tons. Mean draught on trial, 8 feet .02$\frac{7}{16}$ inches.

PERFORMANCE.—TWELVE-HOURS' TRIAL AT TWENTY-FIVE KNOTS.

Steam Pressures. (*Average of hourly observations.*)

	Paulding.	*Drayton.*
Average steam pressure of boilers, pounds gauge	250.0	255.0
at engines, pounds gauge	247.0	255.0
M.H.P. turbine, pounds absolute.	115.0	116.7
I.P. cruising turbine, pounds abs.	255.0	255.0
starboard L.P. turbine, lbs. abs...	23.0	23.3
port L.P. turbine, pounds abs.....	25.0	23.9
on turbine glands, pounds gauge.....	1.0	1.4
auxiliary exhaust, pounds gauge.	4.0	6.7
Vacuum in condensers, inches of mercury, starboard	28.9	28.4
port...	29.2	28.8
Pressure in lubricating-oil system, pounds gauge	5.0	4.5
fuel-oil system, pounds gauge	221.0	262.0

Temperatures. (*Average of hourly observations.*)

Injection, degrees F	62.0	53.6
Discharge, degrees F	83.0	75.8
Feed tank, degrees F	88.0	79.0
water leaving heater, degrees F	221.0	219.0
Engine room, working level, degrees F	105.0	84.0
Firerooms, working level, degrees F	...	75.0
Outside air, degrees F	...	44.0
Smoke stacks, average, degrees F	...	948
Fuel oil to burners, degrees F	74.0	59.0

	Paulding.	*Drayton.*
Oil to oil cooler, degrees F.	92.0	92.2
from oil cooler, degrees F.	87.0	81.2
Water from oil cooler, degrees F.	65.0	56.0
Per cent. moisture, main steam line	3.951	1.1

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, shaft No. 1	668.42	660.4
No. 2	596.41	599.1
No. 3	604.36	618.4
average	623.06	625.9
Pumps, main air, starboard	28.0	30.9
port	26.0	30.5
circulating, starboard	237.0	209.0
port	239.0	217.0
feed, forward	19.0	18.9
after	18.0	16.8
fire and bilge	37.0	21.0
lubricating-oil pump, No. 1	48.0	24.2
No. 2	...	...
auxiliary condenser, air and circulating	...	...
oil-cooler circulating	51.0	43.5
Dynamos, 125 volts, ampères	30.0	35.6
Forced-draft blower engines, forward	1,338.0	1,319.0
after	1,138.0	1,081.0
Speed of ship, in knots per hour	25.308	25.13
Slip of propeller, in per cent. of its own speed, based on mean pitch :		
Shaft No. 1	20.6	20.2
No. 2	16.0	12.1
No. 3	12.2	14.8
Mean	14.6	15.7
Air pressure in firerooms, in inches of water, mean	4.1	3.5

SHAFT HORSEPOWER.

Shaft No. 1	3,290.0	2,744.0
No. 2	1,797.0	2,178.0
No. 3	2,096.0	2,287.0
Total S.H.P. from standardization curve	7,210.0	7,209.0

OIL.

Kind and quality used on trial :		
Paraffin base, specific gravity	0.8789	0.8789
flash point	260.0	260.0
B.T.U. per pound of oil	19,659.0	19,516.0
Pounds used per hour	8,497.0	7,967.8
Knots run per ton of oil	6.67	7.11

DEDUCED DATA.

Square feet of heating surface per S.H.P.	2.01	2.01
Pounds of oil per S.H.P. per hour	1.178	1.15
Cooling surface (main condenser), square feet per S.H.P.	1.332	1.332

WATER.

Pounds per hour, total	105,630.0	104,725
total per hour per S.H.P. of main engines	14.655	14.527
evaporated per pound of oil	12.43	13.14
used by auxiliaries	23,000	23,000*

* Approximate. Auxiliaries exhausting into L.P. turbines.

NOTE.—The percentage of moisture was obtained with carpenter calorimeter. The per cent of moisture given is probably higher than actual, especially for *Paulding*. This is due to readings having been taken too quickly.

DESCRIPTION OF THE HOPKINSON-THRING TORSION METER.

The principle of this apparatus, designed by Professor Hopkinson and Mr. Thring, is a differential one, and consists in the observation of the twist between two adjacent points on the shaft by means of two beams of light projected on to a scale from a fixed and a movable mirror. The beam projected on the scale by the fixed mirror is taken as the zero point, while the beam projected by the movable mirror indicates

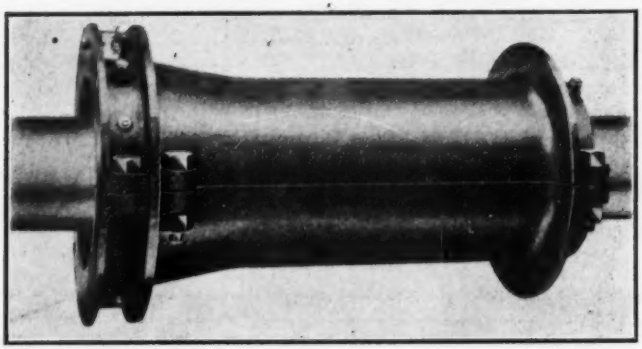


Fig. 1.—TORSION METER MOUNTED COMPLETE ON SHAFT.

the amount of torque on the shaft. Both mirrors revolve with the shaft, but even at moderate speeds the reflections appear as continuous lines of light across the scale.

The Torsion Meter is shown in Fig. 1, mounted complete on a shaft, and the Scale Box in Fig. 2, while a diagrammatic arrangement of the complete apparatus is shown in end elevation and plan in Figs. 3

and 4 respectively. A collar *A*, clamped to the shaft of which the torque has to be measured, is provided with a flange projecting at right angles to the shaft and an extension (Fig. 3).

A sleeve *B* (Fig. 4), provided with a similar flange and extension at one end, is clamped at its further end on to the shaft in such a manner that its flange is close to that on the collar *A*, while its entension overlaps that of the collar *A*, on which it is supported to keep it concentric. Both the collar and sleeve are quite rigid, and it is obvious that when the shaft is twisted by the transmission of power, the flange on the sleeve *B* will move relatively to that on the collar *A*, the movement being equal to that between the two parts of the shaft on which these fittings are clamped. This movement is made visible by one or more systems of torque mirrors mounted between the two flanges, which reflect a beam of light, projected from a lantern, on to a scale divided in a suitable manner on ground glass.

Each system of torque mirrors consists of a mounting, pivoted top

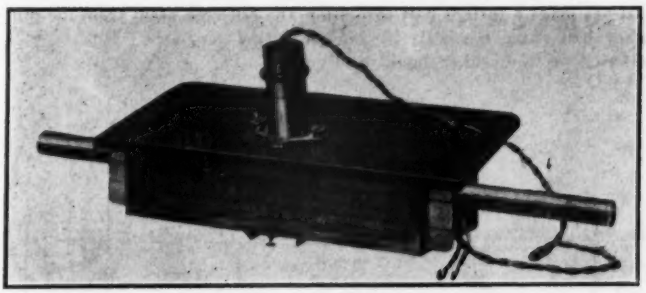


Fig. 2.—Scale Box with Lamp, for Torsion Meter.

and bottom on one or other of the flanges, in which two mirrors are arranged back to back. This mounting is provided with an arm, the end of which is connected by a flat spring to an adjustable stop on the other flange. Any relative movement of the two flanges will turn the torque mirror and thereby cause the beam of light to move on the scale, the deflection produced being directly proportional to the torque applied to the shaft. Hence, if the rigidity of the material and the number of revolutions per minute are known, the H.P. transmitted can be readily calculated.

With the arrangement described, a reflection will be received from each mirror at every half revolution of the shaft; but where the torque varies during a revolution (as with reciprocating engines), a second system of mirrors may be arranged at right angles to the first system, so that four readings can be taken during one revolution; or, if two scales are used, eight readings can be taken.

Fig. 3 shows how the beam of light reflected by the mirror when in its

highest position passes through the upper part of the scale; while the second reflection will occur when the mirror is in the position occupied by the zero mirror, the beam of light passing through the lower part of the scale. The position of the torque mirror in Fig. 4 is such that the

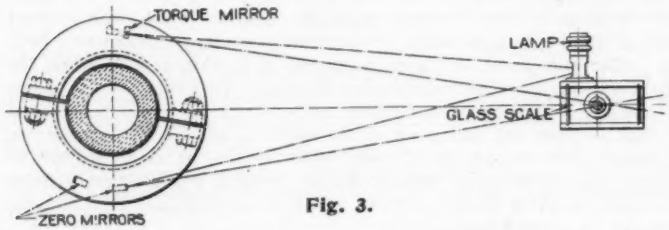


Fig. 3.

reflected beam strikes the scale to the right of the zero line, but when the shaft has made a further half revolution, the reflected beam from the other mirror will strike the scale to the left of the zero line. Obviously the deflection on both sides should be equal.

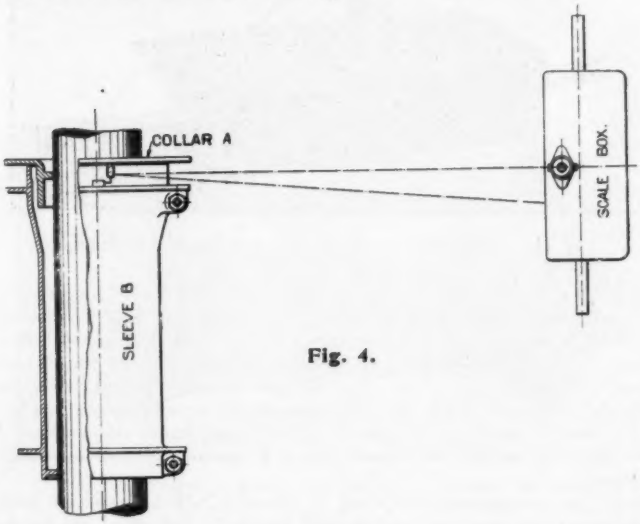


Fig. 4.

The fixed mirror is attached to one of the flanges (in Fig. 4 to the flange of the sleeve *B*). This must be adjusted so that the beam of light reflected from it is received at the same point on the scale as those from the movable mirrors when there is no torque on the shaft. To facilitate the erection and adjustment of the apparatus, the box containing the scale

and carrying the lamp is fitted with trunnions, so that it can be inclined as required.

If the position of the apparatus becomes altered relatively to the scale owing to the warming up of the shaft or from other causes, this is indicated immediately to the observer by an alteration in the position of the zero as reflected by the fixed mirror. Hence, the zero can be adjusted by moving the scale so that its zero coincides with the reflection from the fixed mirror. It will be obvious that it is not necessary to move the scale, as the mean of the two readings will be the same. It will readily be understood that a movement of the torque mirrors can only occur through a relative movement of the two flanges, so that vibration of the shaft or of the ship will not influence the readings.

The constant of the instrument, viz., the factor which, when multiplied or divided into the product of the torsion-meter reading and the revolutions, gives the horsepower, may be calculated within 2 or 3 per cent., if the section of shaft within the instrument is uniform. A direct calibration of the shaft with the instrument in position is recommended before the former is put into the ship. This is effected readily by applying a known twisting couple.

THE CRUISER *DIXIE;* TENDER TO THE ATLANTIC TORPEDO FLEET.

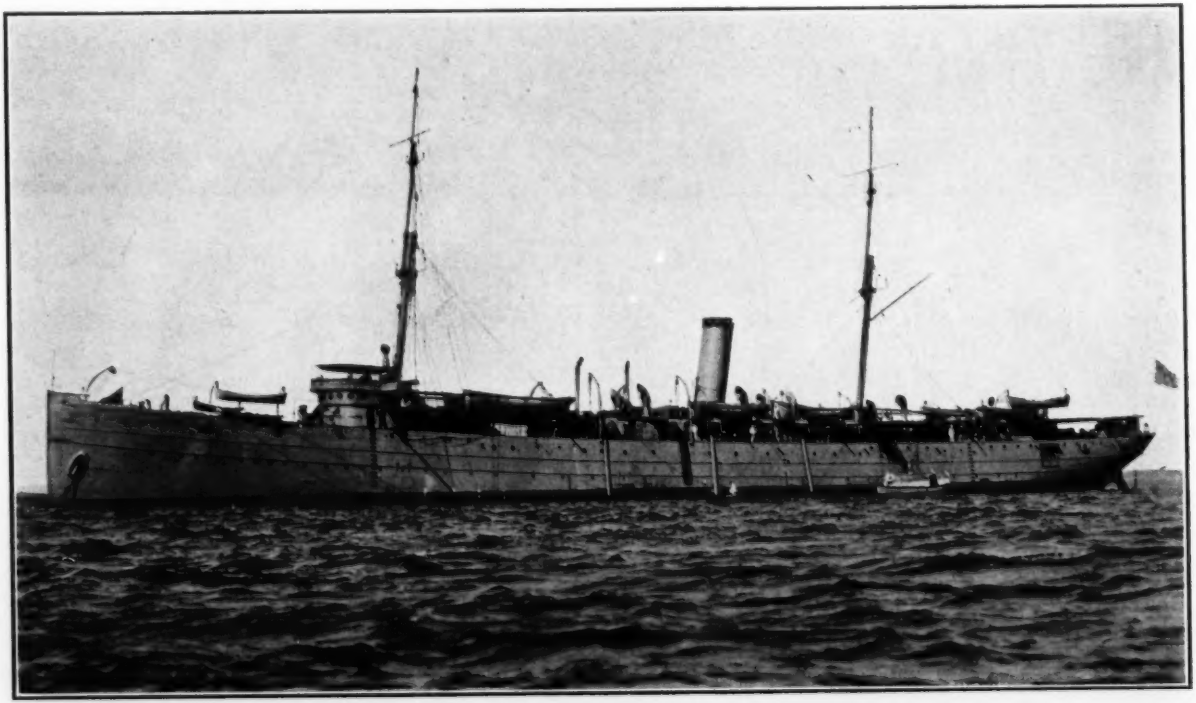
THE LATEST TYPE OF NAVAL AUXILIARY.

By Lieutenant Paul Foley, U. S. Navy, Commanding.

The cruiser *Dixie,* familiar to all shipping men before her acquisition by the Government at the time of the Spanish War as the Morgan Line Steamer *El Rio,* has been refitted for special duty as tender to the Atlantic Torpedo Fleet.

The purpose in view has been to produce a vessel that will satisfy all the needs of a fleet of fifteen or more sea-going destroyers in the matter of repairs, general ship supplies, food, clothing and, to a limited extent, conveniences, for extended periods of time, in order to make it possible for the torpedo fleet to accompany the battleship fleet, and all times to operate with the fleet, without at any time being a handicap to the fleet. Involving an organization of two thousand men, and aggregating over two hundred thousand horsepower, more than the combined power of the heavy fleet, the solution of this ambitious problem has meant in effect the production of a floating navy yard.

The general characteristics of the *Dixie* lend themselves readily to the special requirements of her new duty. She is a vessel of six thousand tons displacement and twenty-two feet maximum draught, having a sustained cruising speed of fifteen knots and a cruising radius at that speed of about six thousand miles. Four hundred feet of parallel side, clear of projections, and protected by heavy vertical wooden fenders, permits the landing alongside without the slightest danger to destroyers which are themselves nearly three hundred feet in



U. S. S. "DIXIE."

length, while the single screw eliminates any possibility of bilging on a projecting propeller blade.

Internally the vessel presents a succession of one magnificent clear deck, and one magnificent storeroom, magazine, or hold after another, all disposed so fortuitously as to suggest in themselves special purposes to which they might be adapted. It is convenient, however, to consider the special features that have been incorporated therein with reference to the purpose they are intended to serve rather than with reference to situation or relative position in the ship. They group themselves naturally under the heads of *Repairs* and *Supplies*.

The machine shop is the biggest and most impressive single feature of the repair department. Located on the after berth deck, a compartment one hundred feet in length, extending the full width of the ship, and having a head room of ten feet, this shop contains twenty-six of the most modern high-grade machine tools, each independent motor-driven, and in their range covering every mechanical operation. The machines are arranged in four parallel lines, fore and aft, with lines of work benches at the outboard sides. There is a large slotted erecting slab and ample floor space for laying off work, swinging torpedoes and similar operations. Air ports on either side placed high enough to be kept open when at anchor illuminate the deck sufficiently by day to render artificial illumination unnecessary. It is provided, however, by a number of cluster lights fitted with porcelain reflectors and arranged in parallel lines so placed with reference to the lines of machines as to reduce shadow casting to a minimum. The cluster of lights are supplemented by drop lights in sufficient number at each machine. Mercury-vapor lights were not installed on account of lack of head room, which is just about sufficient to produce all the glare of that type of light without any of its softening effect.

Photographs or a mere enumeration of tools convey very little idea of the splendid sufficiency of this shop, which chal-

lenges the admiration of all visitors as being one of the most complete and modern small repair plants in the United States. The combination of the utility of the factory with the discipline of the man-of-war is unique in the march of naval progress. The complete machine-tool equipment comprises the following tools: 5 Hendey-Norton lathes of various swings from twelve to twenty-four inches; 1 Harrington gap lathe, 48-inch swing and 18 feet bed; 1 37-inch Niles vertical boring mill; 1 24-inch Woodward and Powell planer, with variable speed drive; 1 No. 4 Dresser universal radial drill; 1 24-inch Cincinnati crank shaper; 1 16-inch Cincinnati crank shaper; 1 Niles horizontal boring mill; 1 Hendey-Norton universal milling machine with all attachments; 1 Bridgeport combination wet and dry grinder; 1 Bridgeport buffing lathe; 1 No. 4 Bignall and Keeler pipe and bolt-cutting and threading machine; 1 Badger State bench punch and shear; 1 Greenard mandrel press; 1 Hisey-Wolf portable tool-post grinder; 1 Cochrane and Bly cold saw; 1 Power hack saw; 1 Hendey-Norton centering machine; 1 28-inch Barnes upright drill.

The tool rooms are very conveniently located just below the machine shop, and are equipped in accordance with the most modern shop practice in this particular.

The blacksmith and coppersmith shops are located at the after end of the machine shop, where the deck is covered with granolithic cement to protect it from fire. In addition to the usual equipment of such shops there are two Mircs oil-fuel forges, a Hauk burner; a tool dressing and forging hammer, operated by air; and bending rolls for sheet-metal work; also a Hilles and Jones, single-end punch and shear, capable of shearing metal up to one inch in thickness.

The compressor room is located on the starboard side of the ship, forward of the machine shop, and is equipped for supplying both high and low-pressure air. All air compressors are steam-driven, and connected to an independent auxiliary condenser in the engine room. Low-pressure air, 125 pounds or less, is furnished by two Westinghouse vertical

units in sufficient volume to operate the steam hammer, oil-fuel forges, brass melting furnace, and also for the various pneumatic hammers, drills, etc., which are in more or less constant use throughout the ship. The air is also piped to the sides of the ship where it becomes available for blowing boiler tubes. High-pressure air is furnished by a large three-stage air compressor, weighing some eighteen thousand pounds, and specially manufactured for this purpose by the Norwalk Iron Works. This compressor will at easy performance compress twelve cubic feet of air, the capacity of the largest torpedoes now manufactured, to 2,500 pounds per square inch in fifteen minutes. High-pressure air connections are also extended to the sides of the ship, and accumulators sufficient to charge four torpedoes are fitted.

The torpedo testing and repair shop, adjoining the compressor room, contains a Rivett precision lathe, with milling attachments, gear cutters and turret; all capable of doing the finest grade of precision work. There is also a Knecht friction sensitive drill, and a buffing lathe, and appurtenances for testing and adjusting the several special mechanisms of all types of torpedoes.

Conveniently placed with reference to the testing room there is an above-water launching tube; while in the magazine space below stowage is provided for a hundred or more spare torpedoes with their war heads.

Positive-acting lifting apparatus is provided for lifting the weapons from the deck of a destroyer, and trolleys for handling them in the ship.

A portable house for testing gyroscopes ashore in isolated places is at hand.

The general object of this extensive torpedo equipment is to make it possible for destroyers to place their torpedoes in condition independently of other occupations, as scouting or steaming trials.

The brass and iron foundry is located on the spar deck amidships, as near the center of motion of the ship as possible.

It contains an iron cupola, capable of melting two thousand pounds of iron per hour; a Mircs oil-fuel crucible furnace; a gasoline preheater, and a complete thermit casting and welding outfit. The discharge from the cupola leads directly into the main smoke pipe of the ship, and is a very convenient method of taking care of the gases and eliminating danger of fire.

A convincing illustration of the possibilities for usefulness of the tender as well as of the sufficiency of its equipment was afforded in the successful welding on board that vessel across a bearing surface of a fractured blower shaft belonging to the destroyer *Reid* and broken during the final acceptance trial of that vessel at Pensacola, Florida. The broken shaft was welded by the thermit process, restored to its original condition of usefulness and refitted in the vessel in seventy-eight hours, at a total cost, including material and labor, of thirty-three dollars. The nearest practicable source of supply for a new shaft would have been Bath, Maine, where the vessel was built.

This operation, which was one of the most exacting ever attempted by the thermit process, was described in the August number of the JOURNAL, accompanied by photographs showing the various steps in the operation.

The shaft was a single-crank solid-forged shaft, turning on a vertical axis at nine hundred revolutions per minute. It was 40 inches long, and 2½ inches in diameter, with an axial hole for internal oil circulation. Complete fracture had occurred about eleven inches from the fan end, across a bearing surface.

The conditions to be satisfied thus were: to recover, and maintain the axial alignment of the two pieces of the hollow shaft; to preserve the over-all dimensions of the shaft mathematically exact, one end being fitted with thrust collars, and the other end being a tapered seat; to avoid burning the metal contiguous to the butts, as the welded metal had subsequently to be turned down to a true bearing surface; and, finally, after

welding, to rebore the axial oil hole, with an additional radial oil hole directly through the weld.

When it is considered that this shaft carried a fan six feet in diameter and turning at nine hundred revolutions per minute in a casing 1/16-inch clearance, it will be appreciated that the recovery of centers must have been accurate to a very precise degree; and also from an inspection of the photograph that while it was comparatively easy to align the two bearing surfaces on each side of the crank with each other, to align the taper bearing with both was a very different matter.

The operation was entirely successful. The *Reid*, after steaming about two thousand miles with the welded shaft, transferred it to the *Flusser*, a sister vessel, in which it has been in constant operation during all the standardization trials, on the Rockland, Provincetown and Delaware Breakwater courses, during the course of which the vessels have frequently been forced to speeds above thirty-three knots.

All shop operations are controlled by a simple but thoroughly efficient card-index and cost-accounting system, by means of which any stranger can at any time ascertain the work in progress or the record of completed work for each vessel, including the names of the mechanics, the machines and materials used and the cost. The record of every machine tool, showing the actual time it has been in operation from day to day and for what purpose, can be ascertained in the time that it takes to read a card.

There is no considerable force of mechanics regularly assigned to the repair force, as might be expected, the idea being that the mechanics of the several vessels will do their own work, using the shop facilities extended by the tender. Much can be said for and against this system. It reduces to a balance between the education of the enlisted personnel on the one hand, and more or less frequent damage to tools on the other. In torpedo-boat repair work, which is almost always "hurry-up" work, and frequently "rush" work, the system which eliminates explanation more or less misunderstood, and

pictures more or less inaccurate by sending the man who knows directly to the machine where he becomes responsible for his own failures, tend to expedite production in no uncertain way. It has a very material disadvantage, however, in that such a system could hardly obtain in time of war, and therefore it should not obtain in time of peace.

The equipment of the supply department is quite as elaborate as that in the repair department, and includes both commissary and general ship's stores. In connection with the former there is a cold-storage plant having a capacity of something like forty thousand rations of fresh meat; while the dry provisions and clothing store rooms are of indefinite size and capacity. In the matter of provisions the vessel is, of course, designed to subsist men under war conditions, that is to supply the necessities rather than the luxuries of life, but the system of centralized supply is so attractive in the matter of convenience, quality and cost, that even on the Atlantic Coast the destroyers subsist almost entirely from the tender. There are two electric bake ovens having a combined capacity of about eight hundred loaves of bread a day. These bake ovens are the latest expression of development in electric cooking appliances, and while similar ones are projected for the new 26,000-ton battleship, the *Dixie* is at the moment the only vessel in the Navy so equipped. Careful tests made under service conditions warrant the statement that the electric oven is equally if not more economical than the coal-burning oven; that is, that for one hundred pounds of coal more bread can be baked in the electric oven and baked in faster time than can be baked in the coal oven. The superiority of the product is, of course, marked, and this combined with the entire absence of dirt, coal dust and all the attendant troubles of coal makes the electric installation very attractive. An electric range has also been installed for test in the officers' galley, but at the moment no information as to its efficiency and reliability is at hand.

The system of "finding" vessels in the matter of general

ship supplies in accordance with a prescribed allowance list or lists which themselves represented the accumulated accretion of years, was, for motives of economy and saving of weight, abandoned about two years ago, and all such vessels are now operated on a money allowance. The commanding officer of the ship is in effect given a letter of credit on the fleet paymaster, within the limits of which he is practically the sole arbiter of the kind and quantity of material drawn. Should the account be overdrawn, however, the supply automatically stops and nothing further can be drawn without authority from the Navy Department. The possibilities for economy and efficiency of administration opened up under such a system are manifestly very great. It necessitates, however, the establishment on the tender of a centralized store house containing a large reserve supply of stores. This centralized store, known in the naval service as the Naval Supply Fund, has reached in the *Dixie* its highest state of development afloat. The stock carried includes every item of ship equipment known in the naval establishment, all housed and administered under the most modern business methods, reducing paper work of every description to a minimum, and maintaining the entire stock, which aggregates over three hundred and fifty thousand dollars, and comprises over five thousand separate items independent of the personality of any individual or combination of individuals. So complete and withal so simple are the accounting methods in use that a balance for quantity or value or both can be struck at any moment.

In conjunction with the naval supply fund there has been issued a catalogue showing the name, naval classification, unit of sale and estimated price of every item of stock, with such explanatory remarks in the case of each item as may be necessary to establish a connection between the unit of sale and the unit of use, which frequently are widely divergent. Sheet metal, for instance, is sold by the pound, but used by the square foot; and for economic administration there must ob-

viously be something to tell the purchaser how many square feet there are in a pound. As a further convenience, catalogue numbers have been assigned to each item of stock. It will be appreciated that ordering by numbers avoids considerable detailed description, especially in connection with bolts and nuts, machine screws and similar material.

In view of the practice of other navies, notably the British, there may exist a doubt as to whether the *Dixie*, on account of her comparatively low speed and light battery is the proper type of ship to have fitted for this duty. The answer seems to be "Yes." If high speed could have been obtained without sacrificing something else, it might have been of some advantage; but a study of the conditions likely to obtain in time of war points to the conclusion that no higher speed than that of the battleship fleet is necessary, as it does not seem within the probabilities of war that a destroyer fleet will get beyond the supporting distance either of its protected base or of the battleship fleet. The proper place for the tender will be either at the base or with the fleet auxiliaries, and not operating in actual company with the destroyers.

Of the battery also—ten three-inch guns, all placed on the spar deck, which may be objected to by some as inadequate—it may be said that qualities in themselves intrinsically desirable are considered of less value in a tender than other qualities, store-room space for example, which they would, if included, necessarily exclude. In the case of the *Dixie*, however, the armament is supplemented by a complete outfit of naval-defense mines used for protecting anchorages, and of counter-mines.

Almost two years have been spent in this great constructive effort, not the least interesting feature of which is the fact that, with the exception of scattered items of minor importance, the equipment has been projected and effected entirely by the enlisted personnel under the supervision of the ship's officers, and have been developed coincident with and in addition to other activities. On the Atlantic coast, where

vessels are never long separated from the great centers of food and material supply, the tremendous military asset embodied in the tender or mobile base is somewhat obscured. It will remain for an extended over-sea cruise to develop to the fullest extent the inherent possibilities, and to emphasize the fact that as she floats to-day the *Dixie* is one of the most useful auxiliaries not only in our own navy, but in any navy.

EXPERIMENTAL-STATION REPORTS.

MONEL METAL.—RESISTANCE TO CORROSIVE EFFECT OF SEA WATER.—CORROSIVE EFFECT ON IRON AND STEEL WHEN IN ELECTRICAL CONTACT IN SEA WATER.

Monel Metal: Corrosion of Iron Electrically Connected with; and Difference of Potential.

Apparatus.—A glass jar filled with Severn river water containing 400 grains chlorine per gallon, a Leeds & Northrup potentiometer, a galvanometer, standard cell and resistance box, a stand for carrying the wires and jar, two wires for connecting test pieces to potentiometer with a switch for short circuiting the current, test pieces, a polished rectangular bar of Monel metal with a wire soldered into one end for the positive electrode and a polished round bar of wrought iron with wire for negative electrode.

Chemical analysis of Monel metal bar:

Nickel, ...	73.9
Copper, ...	25.7
Iron, ...	0.4

Wrought-iron bar, common—Navy specifications.

Connected electrodes to potentiometer after having carefully cleaned, dried and weighed them at 3:50 P. M., January 18, 1910; placed them in Severn river water in glass jars, the immersed area of each electrode being about the same. Kept these two pieces electrically connected, either through potentiometer or short-circuit switch, for ten days, until 3:50 P. M. on June 28th, when removed pieces from water. Measured difference of potential at intervals, as follows:

Date.	Number of readings.	Average difference of potential.
Jan. 18, 4:00 P. M. to 4:30 P. M.	7	.30728
19, 2:30 P. M. to 4:20 P. M.	6	.46602
20, 10:00 A. M. to 4:30 P. M.	8	.51227
21, 9:30 A. M. to 3:30 P. M.	6	.50470
28, 3:10 P. M. to 3:50 P. M.	5	.49905

	Weight of test pieces, grams. Jan. 18th.	Jan. 28th.	Loss in 10 days, grams.
Monel metal, .	110.359	110.359	.0
Wrought iron, .	155.93	155.80	.13

Repolished wrought-iron piece and replaced in same water, immersed the same depth, on February 3d at 2:15 P. M., after having carefully cleaned, dried and weighed it.

Weight on February 3d, grams, ...	134.0778
14th, grams, ...	133.9811
Loss in eleven days, gram, ...	.0967
ten days, gram, ...	.0879
connected to Monel in 10 days, gram, . . .	.1300
alone in 10 days, gram, ...	.0879
due to presence of Monel, in 10 days, gram, .	.0421

On January 21st there was received Monel metal as follows:

	Composition.
4 bars, cast, 30 ins. × 2 ins. × 0½ in.	Cu 24.76, Ni 70.5, Fe 5.24
2 bars, rolled, 60 ins. × 2 ins. × 0½ in.	Cu 25.86, Ni 72.1, Fe 2.04
2 bars, drawn, 60 ins. × 0½ in.	Cu 26.52, Ni 72.6, Fe 0.88

From the rolled bars cut two identical strips, 2 inches × 0½ inch × 0½ inch, marked them A and B and polished them, brushed and weighed them, connected them electrically through a potentiometer, using Severn river water containing 400 grains Cl per gallon as electrolyte, at 12:10, February 12, 1910. At times one piece would be negative and at other times the other piece.

Connected these pieces together with a copper wire, im-

mersed them in Severn river water to $\frac{1}{16}$ inch of upper ends in an agateware boiler and placed boiler on electric heater on February 12, 1910. Each day would boil the water from two to six hours, following it up with additional water from the river to make up for that boiled away.

On February 21st removed these pieces, cleaned and dried them; discovered that on the rolled sides and ends of each piece the surface was a network of fine cracks, as if the surface had checked during rolling. These were not noticed when the pieces were first put in, the dust from the polisher evidently filling them up, making a smooth surface.

	Weight, in grams. Feb. 12th.	Feb. 21st.	Loss in 9 days.
Strip A, ...	58.4512	57.5389	0.9123 gms.
Strip B, ...	60.1609	59.0989	1.062 gms.

Replaced them and boiled water, at times letting it boil dry, until March 2d.

	Weight, in grams. Feb. 21st.	March 2d.	Loss in 9 days.
Strip A, ...	57.5389	57.5253	0.0136 gms.
Strip B, ...	59.0989	59.0923	0.0066 gms.

Replaced them, boiled, as above, until March 11th.

	Weight, in grams. March 2d.	March 11th.	Loss in 9 days.
Strip A, ...	57.5253	57.5103	0.015 gms.
Strip B, ...	59.0923	59.0774	0.0149 gms.

Replaced them, but did not boil water. Weighed again on March 20th.

	Weight, in grams. March 11th.	March 29th.	Loss in 18 days.
Strip A, ...	57.5103	57.5081	0.0022 gms.
Strip B, ...	59.0774	59.0758	0.0016 gms.

Replaced them, boiled them six hours a day for next five days, then allowed to stand, keeping water up to mark by filling up to take place of that evaporated, until May 5th, when removed pieces and weighed them.

	Weight, in grams. March 29th.	May 5th.	Loss in 37 days.
Strip A, ...	57.5081	57.4952	0.0129 gms.
Strip B, ...	59.0758	59.0561	0.0197 gms.

	Weight, in grams. Feb. 12th.	May 5th.	Loss in 82 days.	Per cent. weight.
Strip A, ...	58.4512	57.4952	0.9560	1.634
Strip B, ...	60.1609	59.0561	1.1048	1.836

Of this loss of 1.634 and 1.836 in eighty-two days, a loss of 1.56 per cent. and 1.76 per cent., respectively, occurred in the first nine days. It is believed that the excessive loss is due to the cracks in the rolled side and ends having been filled with dust well worked in by the polisher, and that the action of the water removed the dust, causing this excessive loss of weight on first trial.

The heated pieces, after eighty-two days, were smooth and even, showed no pitting or visible signs of corrosion, the surface being still bright; the only evidence of any corrosion having taken place was the loss of weight and a light grayish scale that was easily brushed off, leaving metal bright under it.

The difference of potential and the current between rolled, cast and drawn Monel of above compositions and samples of steel boiler plate, wrought iron and grate-bar cast iron of Bureau specifications is about two-thirds that between copper and brass and the same respective specimens of the iron family, tried on the same wires, in the same solution, and in every way the tests were similar.

Pieces of cast and rolled Monel metal, copper and brass, numbered Cm1, Rm2 and Cm2, C1, C2, B1 and B2 were placed on pieces of steel boiler plate numbered S3, S4, S10, S1, S2, S5 and S6, respectively, on the bottom of glass fruit jars filled to top with distilled water, except Cm2 on S10, which was filled with clear saturated solution of lime water, pieces of steel being the same distance from surface of water under the same area of opening, all steel pieces being the same size; Monel, copper and brass pieces being of same

breadth and length but of varying thickness, steel pieces S1, S5 and S10 being polished, others having mill scale on two sides, planed on edges.

At the same time placed one steel piece, S1, polished, and one piece, S12, with mill scale in jars alone.

Pieces placed in open jars under distilled water, on April 25, 1910, arranged as follows:

S1	under C1	S1 polished	Distilled water;
S2	under C2	S2 with mill scale	Distilled water;
S3	under Cm1	S3 with mill scale	Distilled water;
S4	under Rm2	S4 with mill scale	Distilled water;
S5	under B1	S5 polished	Distilled water;
S6	under B2	S6 with mill scale	Distilled water;
S10	under Cm2	S10 polished	Clear solution lime water;
S9	alone	S9 polished	Distilled water;
S12	alone	S12 with mill scale	Distilled water.

The loss of S2 under copper, S3 under cast Monel, S4 under rolled Monel, S6 under brass, are comparable, and the increase over that on S12 may be taken as the loss due to presence of metal electronegative to steel on steel with mill scale. The loss on S1 under copper and S5 under brass can be compared with that on S9, showing loss due to metals negative to polished steel.

Piece.	Area, square inches.	Weight, in grams.		Loss per sq. in.	Weight May 24.	Loss per sq. in.
		Mar. 25.	Apr. 24.			
S1	7.14292	96.5029	95.3718	.0184	96.2450	.0178
S2	7.14292	99.2325	99.0406	.0269	98.8831	.0221
S3	7.25750	99.6073	99.4256	.0251	99.2380	.0259
S4	7.27792	100.9482	100.8204	.0175	100.6640	.0214
S5	7.06805	94.9228	94.7879	.0190	94.6508	.0193
S6	7.04339	97.0213	96.8292	.0273	96.6674	.0230
S10	7.30550	98.8737	98.8381	.0049	98.8346	.0005
S9	7.08729	94.5736	94.5736	.0152	94.4511	.0173
S12	7.13728	98.3234	98.3234	.0125	98.1839	.0195

A section of a Monel condenser tube was buffed all over and subjected to a ferroxyl test, showing a positive copper reaction on the outer surface of the tube, with small rust spots like iron rust, and a negative reaction all over the inside surface. Another section of this tube annealed dead soft, and a section hard as received, were placed in a box, and the discharge from the auxiliary condenser circulating pump directed through the inside of tubes, the outside being exposed to river water at high tide and to air at low tide, and have been subjected to this treatment from March 21st to April 21st and from April 29th to May 29th. The inside surface of these tubes shows perfectly bright, while the outside shows small pitted spots, as shown on the ferroxyl, though the annealed tubes show far less pits and corrosion than do the unannealed ones.

Conclusions.

While the results of the various tests varied considerably among themselves, it is evident, as a general proposition, that under the conditions of the tests Monel metal is less corrodible than iron or steel; practically non-corrodible in their presence. Also that the galvanic action arising from the use of this metal in contact with iron or steel in river water was less than when brass or copper were used in the same way, in the ratio of about 3 to 2.

USE OF GRAPHITE AS A CYLINDER LUBRICANT WITH SUPERHEATED STEAM.

Experiments were recently made to determine the suitability of liquid graphite for cylinder lubrication with superheated steam, to the exclusion of cylinder oil.

An engine was fitted up for these tests, all parts being thoroughly overhauled, and valve seat, cylinder, valves and piston all carefully refitted with new parts where necessary. The engine was used to drive the reversed runner of a circulating pump, this arrangement forming a brake.

Four-hour tests were first made, with saturated steam, the

engine cylinder being lubricated on the first test with vaseline, on the second test with cylinder oil, and on the third test with liquid graphite. Indicator cards were taken at the end of each test and data were taken every fifteen minutes.

After the above series of tests was concluded additional tests with superheated steam were made, under as nearly as possible the same conditions. The results of all these tests are given in Table I.

Examination of the above table indicates that, with saturated steam, liquid graphite forms a suitable lubricant for engine cylinders, but the revolutions per minute and the horsepower obtained with practically the same steam pressure as in the other tests were perceptibly less.

Using superheated steam, the graphite proved to be unsuitable, as the tests in each case were terminated by the breaking of some part of the valve gear. The engine worked perfectly with superheated steam, using cylinder oil or vaseline lubricant.

The graphite lubricant was tried in the station gasoline launch, which is fitted with a 4-cycle 2-cylinder engine, making 368 revolutions with forced lubrication. This test had to be given up, as the graphite did not lubricate the cylinder and the engine did not work properly.

TABLE No. I.

EFFECT OF GRAPHITE ON THE CORROSION OF STEEL.

Tests to determine the corrosive effect of graphite and water were made, as follows:

A number of 8-ounce glass-stoppered bottles were used, in each of which 200 c.c. of water was placed. In ten of the bottles distilled water was used, and in ten others sea water with chlorine concentration of 1,000 grains per gallon. To the distilled water in one bottle 20 c.c. of liquid graphite was added. In another bottle 10 c.c., in another 5 c.c., and in a fourth $2\frac{1}{2}$ c.c. The fifth bottle contained only distilled water. These bottles will be designated as those containing sample No. 1.

TABLE No. I.

Date of test.	Lubricant used.	Method of applying lubricant.	Kind of steam.	Engine and steam data.										
				Average for four hour hours' test.					Data when cards were taken.					
				R.p.m.	Steam pressure, by gauge.	Temp. steam, thermometer.	Temp. steam, from table.	Remarks.	R.p.m.	Steam pressure, gauge.	Temp. steam, thermometer.	Temp. steam, by table.	H.P., at r.p.m.	H.P. calculated for 60 r.p.m.
July 20.	Vaseline.	Cylinder and valves wiped out with vaseline before start, 1 teaspoonful every hour.	Saturated.	564.1	180.1	370.8	379.8	Valve, valve seat and cylinder in perfect condition after test.	572	170	...	375.4	9.78	11.28
July 21.	Cylinder oil.	Cylinder and valve wiped out with cylinder oil before start.	do.	569.6	180.2	370 4	379.8	do.	568	180	370	379.8	8.73	10.28
July 22.	Liquid graphite.	Cylinder and valves wiped out before start, 1 quart used.	do.	562.5	182.1	371.8	380.6	Valves, valve chest and cylinder bright and shiny, no evidence of cutting or scratching.	553	180	371	379.8	10.69	13.65
July 26.	9 A. M. to 12 M. Vaseline.	Cylinder and valves wiped out before start. Begun feeding 9:20.	Superheated.	601.3	182.4	401.0	380.6	Test stopped 12 M. Valve seat cut. Scale from superheater coil throughout system.	530	180	...	379.8	7.85	11.49
July 28.	Cylinder oil.	do. ½ pint used.	do.	565.6	168.8	579.8	374.8	Valve chests, cylinder and rings in perfect condition.	590	165	590	373.0	9.79	10.29
July 29.	Noon to 3:37. Liquid graphite	do. 1 pint used.	do.	551.6	160.8	591.1	371.2	Test stopped at 3:37, valve stem broke. Valve seat and cylinder cut badly.	527	160	585	370.9	9.79	14.45
Aug. 9.	Noon to 3:30. Liquid graphite	do.	do.	554.0	160.0	569.8	371.0	Eccentric strap carried away, valve and valve seat cut badly.	555	160	600	371.0	9.68	12 86

TABLE No. II.

Piece No.	Area.	Date put in solution.	Liquid graphite, sample No.	Amount of in 200 c.c. of water, cc.	Kind of water.	Weight in grams on date put into solution.	Weight in grams on Aug. 16, when removed.	Total loss after 35 days, grams.	Loss per square inch area after 30 days, grams.	Loss in per cent. of the loss in water alone.
8080	2.03206	July 12.	1	20.0	Distilled.	20.4929	20.3523	.1406	.0593	299.5
81	2.00511	"	1	10.0	"	20.4263	20.3044	.1219	.0520	262.6
82	2.09503	"	1	5.0	"	21.4372	21.3314	.1058	.0433	218.7
83	2.01809	"	1	2.5	"	20.6986	20.5941	.1045	.0443	223.7
84	2.00400	"	1	0.0	"	20.3337	20.2874	.0463	.0198	100.0
85	2.05287	"	2	20.0	"	20.8892	20.7902	.0990	.0413	223.2
86	2.04009	"	2	10.0	"	20.8002	20.7022	.0980	.0412	222.7
87	1.99449	"	2	5.0	"	20.1731	20.0763	.0968	.0416	224.9
88	1.99570	"	2	2.5	"	20.2124	20.1190	.0934	.0401	216.7
89	2.03210	"	2	0.0	"	20.5614	20.5175	.0439	.0185	100.0
90	2.02377	"	1	20.0	Sea.	21.6410	21.5383	.1027	.0435	368.6
91	2.00107	"	1	10.0	"	20.5473	20.4598	.0885	.0380	322.0
92	2.02950	"	1	5.0	"	20.8394	20.7578	.0816	.0345	292.4
93	2.05660	"	1	2.5	"	21.1842	21.0963	.0875	.0365	309.3
94	2.02224	"	1	0.0	"	20.3230	20.2952	.0278	.0118	100.0*
95	2.04211	"	2	20.0	"	20.9278	20.8289	.0989	.0415	321.7
96	2.05497	"	2	10.0	"	21.1084	21.0076	.1008	.0421	326 4
97	2.00821	"	2	5.0	"	20.5172	20.4198	.0974	.0416	322.5
98	2.02208	"	2	2.5	"	20.6249	20.5359	.0890	.0380	294.6
99	2.04211	"	2	0.0	"	20.7522	20.7214	.0308	.0129	100.0*

* It will be noted that the corrosive effect of sea water appears to be less than that of distilled water. The sea water used was shipped to the station in May, 1910, in wooden casks, from which it was transferred to galvanized-iron storage tanks. It appears to have been robbed of the corrosive effect of water taken direct from the sea.

The same number of bottles was prepared in the same manner, with another sample of the graphite, which will be designated as sample No. 2. These two sets of bottles were duplicated, using sea water instead of distilled water.

In each of the bottles as above prepared a piece of steel boiler plate 1 inch by $\frac{3}{4}$ inch by $\frac{3}{16}$ inch was laid flat on the bottom. In the bottles containing 20 c.c. of graphite solution the piece was covered. In those containing 10 c.c. it was covered to about three-fourths its thickness, and in those containing 5 c.c. it covered the piece to about three-eighths its thickness. Those containing $2\frac{1}{2}$ c.c. covered the bottom only.

The results obtained in these tests are indicated in Table II.

Conclusions from the above Tests.

1st. Graphite is not a good cylinder lubricant for superheated steam.

2d. It is not a good cylinder lubricant for gasoline engines.

3d. Graphite in water doubles and sometimes trebles the corrosive effect of the water, and should not be used on boiler manhole or handhole gaskets nor in any place where it can lodge in pockets in combination with water.

EDITORIAL NOTE.—The conclusions as to the use of graphite based on the above experiments are of great interest. Graphite, in order to make a successful lubricant for engine cylinders of any type, must be introduced in such manner that it will effectually reach the surfaces to be lubricated. The difficulty in so introducing it has been the principal bar to its use. In this connection an article published in the February 23, 1909, number of "Power and the Engineer" is so pertinent that we reproduce it in full:

GRAPHITE AS A LUBRICANT FOR GAS-ENGINE CYLINDERS.

BY WALTER N. DURANT.

Becoming interested in the above subject and having access to a new 6-horsepower horizontal engine, using city gas for fuel, I determined to make some experiments. Finding it impossible to mix graphite and oil and feed

it through the ordinary lubricator, the experiments were confined to feeding the graphite dry through the air intake and continuing the use of cylinder oil through the lubricator. At first about an ounce of graphite was fed through the air intake at short intervals, but after each charge the engine would show increased internal friction; however, it would quickly pick up and then appear to run smoother than before. The quantity of graphite was reduced, and it was soon found that the best results were obtained when the engine was not given more graphite than could be consumed in the cylinder, or about $\frac{1}{12}$ to $\frac{1}{4}$ ounce per horsepower in a ten-hour run. This amount should not be fed all at once, but distributed as evenly as possible throughout the ten hours.

The experimenting extended over a period of four months, and during that time the engine was given some severe tests. The spark plug was always in good condition and never missed fire or became carbonized or short-circuited. The cylinder and valves were frequently examined; the latter were in fine condition and the cylinder did not show a sign of a scratch, but had that smooth, dull appearance which indicates the absence of friction. Unfortunately it was impossible to determine the amount of fuel saved by the use of graphite, as the engine was under a constantly varying load.

Desiring to know what others thought of graphite as a cylinder lubricant, I wrote to forty-five prominent gas-engine manufacturers, asking if they recommended its use in their engine cylinders. The majority of replies stated that the writers had none, or very little personal experience, and declined to express an opinion. The answers containing advice were interesting, but rather conflicting, and no information could be gained from a reply like this:

"It is not customary with us to use graphite in the engine cylinders, although we sometimes use a little."

The following is a little more explicit:

"The great trouble with graphite is to apply it properly, so as not to plug the rings and make them stick. If properly applied, however, graphite is indeed an ideal method of lubrication, but, of course, must be used with oil."

A prominent firm making high-grade auto engines writes:

"We would recommend the use of graphite once in a while in your crank case. Same will do no harm. It has a tendency to close the pores of your cylinder and polish same up so as to increase the compression. It is a good thing."

A large marine gasoline motor manufacturer also says:

"Smear the cylinder walls with it. Once a month is often enough to do this. Of course, in addition the regular amount of oil should be fed through the multiple oiler. Graphite will help to retain good compression."

Another well known gas-engine company writes:

"We use more or less graphite in connection with lubrication, and where properly used much better results can be secured than with lubricating oil alone. If the cylinder has been allowed to cut slightly because of lack of oil there is nothing that will put it in shape so quickly as the use of graphite. Where good flake graphite can be mixed with oil and fed to the cylinder good lubrication is certain."

The manager of a large company making gasoline marine engines writes:

"We consider graphite the best lubricant in the world for gas-engine cylinders. The trouble in using it is in getting it into the cylinder. So far no satisfactory means have been devised. We think so much of the lubricating qualities of graphite in cylinders that we make it a rule thoroughly to coat the inside of every cylinder with it before sending our engines out from the factory. If one of our customers should ask us the question we would tell him to use it by all means if he could get it into the cylinder."

A New York City builder says:

"We think graphite lubrication is very good provided you have the proper means for furnishing the graphite in the required and constant quantity so that it will reach the parts to be lubricated."

The objections to its use were: "Forms lumpy spots on valve seats. Has a tendency to carbonize spark plugs. The expense in using it would overcome the advantages."

Nearly all of the firms which did not recommend the use of graphite pointed out the impossibility of mixing graphite and oil and the certainty of clogging the lubricator if fed in that way. Aside from this the only objection I can see is in using too much at one time in small cylinders.

NOTES.

PROPELLER CORROSION.

The remarkable pitting at the root of the blades of high-speed propellers continues to attract much interest as new cases occur from time to time. The most notable in recent months is that of the forward propellers of the *Mauretania*, which has caused much discussion in marine engineering circles. The latest theory is that the true cause is to be found in the fact that more air reaches the blades of turbine-driven propellers than of the comparatively slow-speed reciprocating-engine screw. There is no lack of evidence to prove that where water carries much air its corrosive properties are enhanced. It is, for example, a familiar experience that the bottoms of iron vessels covered with a very thin layer of water rust more quickly than if the depth of water is considerable, and the corrosion of condensers has in several cases been directly traced to the fact that the circulating water was broken up in contact with air—as, for example, by discharging roughly into a tank—before reaching the condenser. On the other hand, however, it must be remembered that the turbine-driven propellers which are causing trouble do not in many cases revolve at a higher speed than the propellers of former high-speed craft, such as destroyers and torpedo boats, so that the explanation does not appear to be entirely satisfactory. That air does find its way to the center of propellers has been proved beyond doubt by many photographs of models, but before it can get there it must pass over the rest of the blade surface, and some fuller explanation of why the pitting should be nearly always concentrated at the roots of the blades is still required. We have suggested on previous occasions that the bending which almost certainly takes place may be a contributory cause, and it has been suggested also

that the form given to the root in order to diminish this bending is such as to cause disturbance in the water, and so liberate the air actually in contact with the metal. Under such conditions air is most active. If this explanation is correct, the fact that the corrosion nearly always takes the form of pitting may be ascribed to the fact that the roughness caused in the surface by a patch of initial corrosion would tend to hold the air, and so both increase and localize the effect. The whole matter requires the closest investigation. In the *Mauretania* the curious fact has been observed that the after propellers do not suffer to anything like the same extent as the forward propellers. This has been attributed to the fact that most of the air is knocked out of the water by the forward propellers, and that the aft propellers receive water almost free from air. Unfortunately, the conditions for both pairs are not the same, the forward pair not only being nearer the ship, but of a different form, the tips being bent backwards in order to increase the clearance between them and the hull, the smallness of which was the cause of the early vibration in this vessel. This shape may actually assist in drawing air into the propeller, and so cause greater corrosion. We suggest that some of the colleges which are provided with hydraulic plant might do worse than conduct an inquiry on this subject. By using highly polished steel screws—with plenty of manganese to increase the electrolytic effect—indications would be rapidly given of the areas which are most readily attacked with different forms of blades, etc. It would be interesting to know if paddle wheels, which must always be in contact with highly aerated water, are particularly subject to corrosion.—"The Engineer."

NEW SHIPBUILDING PLANT.

Bids have been asked for the erection of the new shipbuilding plant of the New London Ship and Engine Company at Groton, Conn., and the award of the contract it is said will be

made soon. At this yard as soon as possible two marine internal-combustion engines are to be built of the Maschinen Fabrik Augsburg-Nurnburg and Blohm & Voss type, the American rights for which are owned by the Electric Boat Company, which will be part of the new establishment. This is the type of engine that will be placed in the new Hamburg-American liners contracted for some time ago. The plant of the Electric Boat Company now in the Fore River Shipbuilding Company's yard at Quincy, Mass., is to be removed to the new yard. Those at the head of this enterprize are T. A. Scott, of New London, W. H. Reeves, Henry Bond, and Capt. Frank T. Cables, superintendent of the Electric Boat Company at Quincy.—" Marine Journal."

LEAVING TOOLS AROUND.

A naval engineer has been reduced several numbers because he left a monkey wrench in one of the cylinders of the ship where he was employed. It was not because the wrench was of much value, and because his Uncle Samuel had to go out and purchase another, but because the wrench was in the way. It wasn't needed in the cylinder. It was foreign matter, not necessary to the running of the engine or to the speed of the boat. It added nothing to the fighting qualities of the craft, and that is what the Government is after principally. In fact, the wrench in the cylinder was a detriment and a hindrance to the smooth working qualities of the boat's machinery. There was no room for it in the cylinder, and when the great engine was started up something, of course, had to go. The monkey wrench being a hard nut to crack, it naturally followed that the cylinder did all the cracking. The cylinder came to grief, and in the common run of things naval the engineer followed the same course.

We are sorry for the engineer, but glad the naval board has meted out this just punishment. It makes us feel more at ease

over the safety of our country not to have monkey wrenches in our cylinders and, to bring the lesson ashore, it will help some when we are undergoing repairs at the hands of the dentist or the surgeon who is using wrenches and scissors and chisels, etc., in the region of our own cylinders. It is a matter of common knowledge that all sorts of trinkets, such as lances, needles and shears, have been sewed up in the human system following a successful operation. Teeth have been found in the stomach long after a siege of extraction and distraction at the dentist's.

A pair of scissors left in a man's interior is of no more use to him than a monkey wrench left in a cylinder. The surgeon may not miss the scissors, but the patient misses the room he had before the scissors usurped it. Space in a cylinder or in a man's anatomy is valuable, and the engineer or surgeon who uses such space for cold storage for tools ought to be reduced, not only in numbers, but to a pulp.—" Boston Herald."

At a banquet given last June by the graduating mechanical engineers of the University of Illinois, Professors Thorpe and Goodenough of that institution spoke a few parting words on the successful engineer, using as their text the time-honored formula for indicated horsepower, which is known to almost every engineer.

$$\frac{PLAN}{33{,}000} = \text{I.H.P.}$$

One of our contributors was so impressed with their treatment of the subject that he forwarded the substance of what was said to " Power," and in slightly modified form we are repeating the gist of their remarks.

Although the talk was delivered to technical men, it applies equally well to the practical worker, and in fact to all men of any profession or trade.

To begin with, the " 33,000" in the usual horsepower for-

mula is a constant and therefore may be represented by a symbol.

For our purpose $1 \div 33{,}000$ is best represented by K, making the formula read:

$$P\,L\,A\,N\,K = I.H.P.$$

As P comes first it is likely to be of great importance, which is true of *Patience*. No characteristic is more helpful under trying circumstances, and nowhere is the exercise of patience more necessary than in the engine and boiler rooms. The long hours of service, the Sunday jobs and the repair work common to even the best of plants, demand that this one quality be developed to an unusual degree.

P L A N K = I.H.P.

P = Patience **A** = Accuracy
L = Loyalty **N** = Nobility
K = Knowledge

I.H.P. = Important High Position

L fittingly comes next; it ought to be prominent, for *Loyalty* to one's employer as well as to one's higher self is the essence of success. The good of the firm demands that all pull together. No minor jealousies or personal affairs should ever be given precedence, for, even from a selfish motive, loyalty to the firm means success to the individual.

Yet no engineer is a real success unless he has the next quality, *Accuracy*, denoted by *A*. There are few occupations in which accuracy is necessary to such an extent as that of the engineer.

As a fitting addition to the list of good qualities we have *N*

for *Nobility*. Be noble! Treat your fellowmen as a man among men. Live as you would like to see men live—as you would have your son live.

Last, but by no means least, there is *K* for the indispensable *Knowledge* that enables the engineer to make the best use of the four qualities mentioned.

Turning to the other side of the equation, there are the symbols, *I.H.P.*, the equivalent of the first member. The result of such a combination of qualities can only be *Important High Position.*

You have then Patience, amplified by Loyalty, strengthened by Accuracy and magnified by Nobility; all these multiplying your Knowledge, and the whole resulting in Important High Position, or Success.

Adopt this "plank" and see how near the top it will land you.

Think it over.—"Power and The Engineer."

SHIPS.

UNITED STATES.

Bids were opened at the Navy Department on November 8, 1910, for the construction of six torpedo-boat destroyers (*Nos. 37–42*) authorized by the naval appropriation act, approved June 24, 1910. The important dimensions of these vessels are the same as for destroyers recently built for the United States Navy.

Bids were invited under three classes as follows:

Class 1.—Government design, hull and machinery.

Class 2.—Government design, hull and machinery, with speed-reduction gearing fitted between cruising and L.P. turbines.

Class 3.—Government design, hull and equipment, bidder's design machinery.

Class.	No. of vessels.	Speed.	Time to complete.	Price.	Water consumption.		Remarks.
				Fore River Shipbuilding Company.			
		knots.	*months.*		*knots.*	*pounds.*	
3	1	29½	24	$648,700	29½	5,800	63-inch, 18-stage, Curtis reversible marine turbine.
					25	4,070	
					20	2,710	
					16	2,110	
					12	1,820	
				Bath Iron Works.			
1	1	30	24	$669,750			
1	2	...	24	654,500	29½	6,150	
					25	4,520	
					20	3,080	
					16	2,180	
					12	1,540	
3	2	29½	24	630,500	29½	5,500	
					25	4,280	
					20	2,850	
					16	1,820	
					12	1,460	
3	1	29½	24	655,500			

Class.	No. of vessels.	Speed.	Time to complete.	Price.	Water consumption.		Remarks.
				New York Shipbuilding Company.			
		knots.	*months.*		*knots.*	*pounds.*	
1	1	29½	24	$645,000	29½	6,252	
					25	4,224	
					20	2,921	
					16	2,078	
					12	1,787	If tried Delaware Breakwater, $5,000 less.
3	1	29½	24	670,000	29½	5,800	
					25	4,070	
					20	2,710	
					16	2,110	
					12	1,820	
				Union Iron Works.			
1	1	29½	24	$721,000			
1	2	29½	23	704,000	29½	6,200	
			24	...	25	4,600	
					20	3,100	
					16	2,200	
					12	1,600	
2	1	29½	24	721,000			
2	2	29½	23	704,000			
			24				
				Maryland Steel Company.			
1	2	29½	24	$685,000	29½	6,252	
					25	4,224	
					20	2,921	
					16	2,078	
					12	1,787	
3	2	29½	24	742,000	29½	5,459	
					25	3,468	
					20	2,231	
					16	1,868	
					12	1,312	
				Newport News Shipbuilding Company.			
1	1	29½	24	$635,000	29½	6,300	
					25	5,200	If tried Delaware Breakwater, $630,500.
					20	2,800	
					16	2,100	
					12	1,400	... Considers 12 tons additional machinery weight necessary.
				William Cramp & Sons Shipbuilding Company.			
1	1	29½	24	$658,800	29½	6,150	If standardization trial Delaware Breakwater, and other trials near Delaware Capes, deduct $4,800.
					25	4,175	
					20	2,800	
					16	2,300	
					12	1,760	

U. S. S. Cyclops. — A successful 48-hours' sea trial of this vessel was completed on October 28, 1910, and all guarantees under the contract were fully met. This vessel, built by the William Cramp & Sons' Ship and Engine Building Company, at a contract price of $822,500, is one of two fleet colliers authorized by the Naval Appropriation Act, approved May 13, 1908. The hull is of steel throughout, and plans and specifications conform to rules of the American Bureau of Shipping.

The guaranteed average speed of the vessel on a continuous 48-hours' trial in the open sea was 14 knots, when carrying 12,500 tons of bunker and cargo coal, and with all equipment complete, including 100 tons of reserve-feed water, 20 tons of drinking water and 130 tons of stores for the crew. It was a further stipulation of the contract that the coal consumption for all purposes during this trial should not exceed 1.8 pounds per I.H.P. per hour of the main propelling engines.

The important particulars of hull and machinery are given below:

HULL.

Length between perpendiculars, feet and inches	520–0
over all, feet and inches	542–0
on L.W.L., feet and inches	528–0
Beam, extreme, feet and inches	65–2¼
molded, feet and inches	65–0
Draught (load), mean, feet and inches	27–6
Displacement on load draught, tons	19,270
per inch, at L.W.L, tons	64.1
Area immersed, center line, load draught, square feet	1,750
L.W.L., plane, load draught, square feet	26,905
Wetted surface, load draught, square feet	49,450
Coefficient of fineness, block	.726
center line	.984
L.W.L	.796

MACHINERY.

Engine:

Type, twin screw, 3-cylinder, vertical, triple-expansion, with cranks at 120 degrees.

Diameter of cylinders:	
H.P., inches	27½
I.P., inches	46
L.P., inches	76
Stroke inches	48
R.p.m	94
I.H.P. (designed), each engine	3,600

Engine:	*H.P.*	*I.P.*	*L.P.*
Number and type of valves	1-piston.	2-piston.	2-piston.
Diameter of valves, inches	14	14	26
Travel of valves, inches	8	8	9
Length of connecting rod between centers, inches			108
Ratio of connecting rod to crank			4.5

	Length.	*Diameter.*
Crank shaft (solid, forged steel), feet and inches	23–03¾	0–14¾
Crank pins (solid, forged steel), feet and inches	0–15½	0–14¾
Thrust shaft (solid, forged steel), feet and inches	16–03	0–14¾
Stern-tube shaft (solid, forged steel), feet and inches	28–03½	0–15½
Propeller shaft (solid, forged steel), feet and inches	33–0	0–15½

Condenser:

Type, surface, one for each main engine.	
Number of tubes (each condenser)	2,447
Diameter of tubes, outside, inch	0¾
Length of tubes, feet and inches	12–8
Thickness of tubes, B.W.G	16
Cooling surface (each condenser), square feet	5,004

Propellers:

Number and material: two, built-up, cast-steel hubs, manganese-bronze blades:	
Number of blades, each propeller	3
Diameter, feet and inches	16–6
Pitch, adjustable, feet and inches, from	17–0 to 19–6
Pitch, as set, mean, feet and inches	18–0
Area, projected, square feet	64.8
helicoidal, square feet	76.8
disc, square feet	202.8

Boilers, main:

Number and type, three double-ended, cylindrical, return, fire-tube:	
Designed working pressure, pounds	190
Diameter, feet and inches	16–5 11/12
Length, feet and inches	23–0¾
Grates, length, feet	5⅝
width, inches	3½
Number of furnaces, each boiler	8
Total grate surface, all boilers, square feet	450
heating surface, all boilers, square feet	19,379
Diameter of tubes, outside, inches	2½
Length of tubes, inches	93¾
Number of tubes, one boiler	1,076
smokepipes	2
Area of smokepipes (each), square feet	43
Height of smokepipes above grates, feet	97.44
Kind of forced draft	Closed ash-pit.

The *Cyclops* was standardized over the Delaware Breakwater course on October 25 and 26, 1910, just prior to beginning the 48-hours' sea test, and from data thus obtained it was found that a mean of 88.37 revolutions per minute of both shafts was required to give a speed of 14 knots per hour. Mean draught during these trials was 27 feet 4¼ inches, corresponding to a displacement of 19,146 tons. The following is from standardization runs:

Speed, knots.	*R.p.m.*	*I.H.P.*
7.998	51.25	1,243
9.784	62.91	2,246
12.16	76.74	4,046
14.089	88.95	5,981
15.485	97.97	8,312

The 48-hour trial was begun at 2:30 P. M., October 26, ending at the same hour October 28, 1910. The course was from Five Fathom Bank Lightship to Nantucket New South Shoal Lightship, thence to the latitude of Cape Henry. At the start the weather was clear and pleasant with smooth sea. During the night of October 26 the wind shifted from N.W. by W. to S. by W., and increased to a fresh breeze with a moderate sea from the same direction. On October 27 there was a moderate gale from S.W. by W. and a rough sea. From noon of that date until near the end of the trial the *Cyclops* steamed into a rough head sea, which occasionally broke over the forecastle.

A synopsis of the important data recorded during this trial is given below:

	Starboard.	*Port.*
Maximum average revolutions per minute for 60-minute period	95.42	95.53
Average revolutions per minute for 48 hours	92.109	92.336
Mean revolutions per minute, both engines	92.261	
Maximum steam pressure at boilers, pounds	197.0	
Average steam pressure at boilers, pounds	192.0	
Maximum air pressure in ashpit of boilers	1.0	
Average air pressure in ash-pit of boilers	.41	
vacuum, inches	26.4	27.7

Collective I.H.P. of main engines	6,705
Average pounds of coal used per hour	9,954
Draught at beginning of trial, (forward), ft. and ins	26–11
(aft), feet and inches	28–00¾
Mean draught at beginning of trial, feet and inches	27–05¾
Corresponding displacement, tons	19,254
Draught at end of trial (forward), feet and inches	26–10¼
(aft), feet and inches	27–11¼
Mean draught at end of trial, feet and inches	27–04¾
Corresponding displacement, tons	19,108
at middle of trial, tons	*19,095
Average speed, knots per hour	14.61
Power of main engines:	
Average revolutions (eight sets of cards)	92.336
mean effective pressures:	
H.P.	90.1
I.P.	31.8
L.P.	10.5
Average indicated horsepower from cards:	
H.P.	1,160.0
I.P.	1,163.0
L.P.	1,077.0
Total (one engine)	3,400.0
Average revolutions or double strokes of auxiliaries:	
Main air pumps	Attached.
circulating pumps	158.0
feed pumps	22.8
Forced-draft blowers	278.0
Sanitary pump	22.0
Ice machine	107.0
Dynamo engine (123 volts), ampères	80.3

DEDUCED DATA.

I.H.P. (main engines only) per square foot of grate surface	14.9
Pounds of coal per I.H.P. per hour (main engines only)	1.48
square foot of grate	22.12
Slip of propellers in per cent. of their speed	10.83

*165 tons of salt water added to after-peak tank at 9 P. M., October 27.

ARGENTINA.

It is reported that the Argentine Republic has in contemplation ordering a third battleship of larger dimensions than the two now building in the United States. This vessel will be designed to meet the design of the *Rio de Janeiro,* now building in England for Brazil.

AUSTRIA-HUNGARY.

There is under construction at Fiume, a vessel built for service as a tender for submarines. The vessel has the following characteristics: Length, 64.2 meters; beam, 10 meters; displacement, about 1,500 tons; engines, 2,500 horsepower; speed, 15 knots; bunker capacity, 200 tons; 2 steam winches, with a power of 15 tons each; compressed-air pumps with reservoirs, with stores necessary for torpedo vessels and submarines, diving apparatus and repair shops.—"La Vie Maritime."

BRAZIL.

The following table gives the results of trials of the ten destroyers recently completed for Brazil by Messrs. Yarrow & Co., of Glasgow:

Results of Official Three-Hours' Full-Power Trials.

Name of boat.	Speed.	Revolutions.	I.H.P.	Air pressure in stokehold.
	knots.			*inches.*
Para, ...	27.259	333.04	7,014	1.22
Piauhy, . . .	27.211	330.70	6,563	1.53
Amazonas, . .	27.178	336.72	6,898	1.64
Matto Grosso, .	27.160	338.11	7,403	1.63
Rio Grande do Norte,	27.275	341.70	7,778	1.90
Parahyba, . . .	27.290	335.44	6,700	1.26
Alagôas, . . .	27.253	337.70	7,403	1.51
Santa Catharina,	27.305	337.50	6,982	1.55
Parana, . . .	28.736	354.40	8,877	2.43
Sergipe, . . .	27.605	364.50	8,554	1.66

Each of these vessels is of the following particulars:

Length between perpendiculars, 240 feet; beam, 23 feet 6 inches; depth, 14 feet, and draught 17 feet 6 inches; displacement, 650 tons. The ratio of length to beam is 10.2 to 1. The

vessels are notable for several features, prominent among them being the adequacy of the living quarters, the application of refrigerating installation for the purpose of cooling the magazines, and the circulation of cold air in the ward room, cabins, and other parts. Each of the vessels carries a large motor pinnace. The Marconi wireless apparatus has been installed, the aerial lines passing from the mast to a special jigger aft. High-tensile steel is used in the parts of the hulls where longitudinal strength is important, and the rivets used are of the same strength per square inch as the plates they connect. The ships are divided into ten watertight compartments by bulkheads from the outer bottom plating to the upper deck.

The vessels are fitted with two sets of four-cylinder triple-expansion engines, balanced on the Yarrow-Schlick-Tweedy system, supplied with steam from two double-ended Yarrow boilers. The firing of these boilers from both ends is a unique feature; the total grate area is 267.5 square feet, and the heating surface 14,980 square feet, the ratio of heating surface to grate area being 56 to 1.

In the maneuvering trials the diameter of the turning circle at full speed was about 375 yards, and at two-thirds speed about 330 yards. In the stopping and starting trials the boats began to move astern from full speed ahead in thirty seconds from the time the signal was given by the telegraph, while they began to move ahead from full speed astern in twenty-five seconds from the time of moving the telegraph. The coal-consumption trials showed that at 14 knots speed the radius of action was 3,690 nautical miles; the bunker capacity is 140 tons.—"Engineering."

CHILI.

It is understood that the principal warship builders of this country and abroad have received an invitation from the Chilian Government to tender for the construction of two bat-

tleships. These ships, it is said, will closely resemble the English super-Dreadnoughts of the *Orion* type, and are to be of about 24,000 tons displacement, with a speed of 23 knots. The engines are to be of the Parsons marine turbine type, and the armament will consist of ten 12-inch or 13.5-inch guns, together with an auxiliary armament of 4.7-inch guns in batteries. There will also be the usual complement of torpedo tubes and smaller guns. It is expected that there will be keen competition for the vessels.—" Page's Weekly."

CHINA.

Prince Tsai, who is the head of the Chinese navy, has been visiting this country with the avowed object of studying the American navy and deciding on the possible purchase of ships. Although bids for whatever ships China may decide to buy will probably be made in the open market, yet it is known that the Celestial officials have a decided leaning in favor of the United States, induced by the liberal treatment which we have always accorded that country. The progressive element, especially, is very friendly towards this country, and the State Department has always been willing to foster this feeling. It is said that there is an excellent chance for American shipbuilding firms to secure liberal contracts in case China decides to build new vessels.

Prince Tsai was accompanied by Admiral Sah, an accomplished Chinese naval officer, who received his naval education in England and commanded one of the high-powered cruisers purchased by China in England about ten years ago.

FRANCE.

Submersible Cruisers.—While French naval experts have lately been deprecating any attempt to exaggerate the value of submersible and submarine boats, they are yet fully alive to

the importance of the work that may be done by these craft. In a country where public opinion has exercised so much influence upon naval questions in the past, it has been found necessary to explode the old fallacy that the submarine has enormously depreciated the value of the battleship. As the Government intends to carry through the important program of naval constructions, in which the battleships of 23,000 tons constitute the main feature, an effort has been made to impress upon the public just what the submersible and submarine are capable of doing, and, judging from the performances of these craft at the recent maneuvers, it has to be admitted that they possess considerable limitations. Nevertheless, it is believed that the submersible, if given a sufficient range of action, will prove very valuable in blockading an enemy's ports and, perhaps, intercepting an enemy's squadron, and the whole aim of the French marine in designing the new submersibles is to increase the range as much as possible. In a word, the idea is to develop a type of "submersible cruiser." As a means of ascertaining exactly what these craft can do, it has been decided to send the *Archimède* from Cherbourg to Toulon on a single run without putting in at any port on the way. The *Archimède* was launched a year ago. Having a length of 60 m. and a beam of 5 m., it displaces 800 tons with its water tanks empty. It has two triple-expansion engines and two propellers, which, when the vessel is submerged, are driven by electric motors. There are seven torpedo tubes. The vessel has a crew of thirty-two, and three officers. The forthcoming run of the *Archimède* is regarded as of great importance, since upon it depends the construction of the new submersibles which, in the opinion of French experts, are destined to carry out effective operations at a considerable distance from their base.

The work of refitting the hull of the French submarine *Pluviose*, which has been towed from Calais to Cherbourg Dockyard, is being rapidly pushed forward in order that the

authorities may have an early opportunity of deciding whether she can be recommissioned for service. Investigation has proved that, at the moment of the collision, short circuits broke out all over the vessel, especially in the central post, where all the cables were found fused, the woodwork of the switchboard charred, and the ebonite twisted or broken. Although the rapid invasion of the water must have prevented the fire from being of any considerable duration, the powerful heat may quite possibly have asphyxiated the men in the central post, as well as the officer in charge of the periscope. It is even suggested that the dense smoke might have been the cause of the opening of the forward manhole. It is considered probable that the vessel was not under control at the moment of the collision with the mail steamer. All operations causing the submarine to rise had been carried out. If the fire broke out before the collision, it would have been this which rendered the crew powerless to prevent the castastrophe.—"The Engineer."

Salvage Vessel for Submarines.—Concerning the *Pluviose* disaster, it has been stated that plans had been approved for a salvage vessel to cost about £12,000, but that the increase in size of the future submarine had delayed its immediate construction, and, in the meantime, a lighter, provided with mechanical means for lifting submerged vessels, would be built.

Referring to the safety helmets that are used in the British navy, Admiral de Lapeyrère said that the results of the trials which had taken place at Cherbourg are not such as to inspire confidence in their efficacy. He called attention to the fact that while there had been 30,000 descents since 1900, only three serious accidents had occurred, and that, while there will always be naval risks, no seaman worthy of the name would be deterred by them from his duties.

Old Vessels.—The French Minister of Marine has struck at the root of what has long been a serious evil. He has consigned some thirty old vessels to the scrap heap. These old

vessels, while in commission, employ a large number of men and officers, and at the same time the modern battleships of the fleet are short at least one hundred men each and the flotillas are now deficient at least one-third of their normal force.

By condemning these useless ships, at least two thousand men have been added to the personnel of the newer vessels.

The French Maneuvers.—A writer in the "Moniteur de la Flotte" makes a few criticisms concerning the recent French maneuvers in the Mediterranean. The scheme represented the situation that would exist if the French fleet were opposed to the fleets of Italy and Austria seeking to unite. The blue squadron represented the French, and the red its enemies. The red fleet based its operations on the intention of deceiving its adversary and did not attempt to pass the Strait of Bonifacio on account of the submarines.—"Page's Weekly."

GERMANY.

Zoelly Turbines in the German Navy.—We are informed that, in consequence of the excellent results obtained in destroyers fitted with the Zoelly marine turbine, the German naval authorities have ordered from the Germania Yard, Kiel, six more boats thus equipped. Official data as to the trials have not been published, but the figures given in the following report, translated from the "Zeitschrift des Vereines Deutscher Ingenieure" for March 26 last, are understood to be authentic. The torpedo boat *G 173*, built by Messrs. Krupp & Co., of Germaniawerft Kiel, carried out her trials on the measured mile in Eckenford Bay. This boat has a displacement of 650 tons, and is propelled by Zoelly turbines, designed by Messrs. Escher Wyss & Co., the total power being 16,000 horsepower. On trial, in about 25 meters of water, and in a slight wind, this boat reached a speed of 33.35 knots, the shaft

power at this speed being 8,200 horsepower in each shaft at about 690 revolutions. *G 173* is the first German warship fitted with Zoelly turbines.

The two new German cruisers Coln and *Augsburg* will make their trials next autumn. They are sister ships of 4,350 tons displacement, 426½ feet length, 46 feet greatest beam, and a draught of 16½ feet. The crew of each will number 363, the speed will be over 25 knots, and the coal capacity 900 tons. The trials of the two ships are being looked forward to with interest in engineering circles, as they are to be tested against the cruisers *Mainz* and *Kolberg*, in order to ascertain the merits of the different turbine systems in use in Germany. The *Coln's* turbines are Zoelly, the *Augsburg's* Parsons, the *Mainz's* Curtis, and the *Kolberg's* Schichau. The *Mainz* averaged 25 knots on her trials, but on one occasion reached 27.5 on the measured mile.—"The Engineer."

GREAT BRITAIN.

H.M.S. Lion.—An event of importance on August 6th was the launch of H.M.S. *Lion*, the largest cruiser in the British navy. No official statement is forthcoming regarding her size or possibilities, but there is good reason for presuming that the following figures are approximately correct: Length, 700 feet; beam, 88½ feet; indicated horsepower, 70,000; speed, 30 knots; displacement, 26,000 tons; main armament, eight 13.5-inch; coal, 3,500 tons; turrets, four central. Although the *Lion* and her sister ship, the *Princess Royal*, are officially described as cruisers, in many respects they are superior to the most powerful battleships afloat. Swifter than either the *Mauretania* or *Lusitania*, combined with greater gun power than any previous vessel of the *Dreadnought* class, they have been well described as the most remarkable engines of war the British naval authorities have ever conceived. In fact, as the

"Birmingham Post" says, they are Lord Fisher's final word in naval design, aided by the constructive genius of Sir Philip Watts, the head of the Royal Corps of Naval Constructors. The remarkable fighting power of the vessel can be appreciated when it is stated that her eight 13.5-inch guns, the first of the new type to be mounted, will discharge a projectile of 1,250 pounds with sufficient force to penetrate 22 inches of armor at a distance of 5,000 yards. The new battleship was laid down on November 29th last year. Her launching weight was 8,057 tons. Speaking of the *Lion* as she was being finally prepared to be put into the water, a dockyard authority is quoted as saying that "in length, beam, displacement, main and secondary armament, torpedo equipment, armor protection, in offensive and defensive qualities generally, and also in regard to a number of minor details of internal construction, the *Lion* is enormously superior to any cruiser built or projected."—"Page's Weekly."

It is not without interest to consider the effective horsepower required to propel ships of different lengths at a given speed, because there is thereby established not only the advantage of the greater length of the *Lion,* but the difficulties under which Admiralty designers have hitherto worked, in comparison with the relative freedom of the designer of merchant ships. It has been put on record, for instance, that with the limitations imposed by other considerations—such as passenger accommodation, etc.—the *Lusitania* and *Mauretania* could not have been built on a restricted length of 700 feet, instead of 760 feet, to cross the Atlantic at the speeds they have attained. The power, if the lesser dimension had been adopted, would have gone up at least 10 per cent., and could not have been installed in the space available. The *Invincibles* are 530 feet, the *Indefatigable* 555 feet, and the *Lion* 700 feet between perpendiculars. The displacement has gone up from 17,250 tons to 26,350 tons. Assuming a speed of 27 knots for all three vessels, the effective horsepower per ton of displacement in the case of the *Invincible* is 1.7, in the

Indefatigable 1.58, and in the *Lion* 1.43. It will thus be evident that the increased length, with the same weight of machinery and the same power, adds materially to speed. The *Lion*, however, will probably materially exceed 27 knots, and the gain will be still more marked.

It is only right to say that such high speed has been rendered possible only by the invention of the turbine, because the steam consumption for this type of machinery is so much less than with the best reciprocating engines yet constructed for naval work. The steam-generating plant, and the weight and space required for its accommodation, is thus enormously decreased. It is scarcely necessary to point to the advantage of the decrease in space, as well as in weight, in view of what we have said regarding the arrangement of the armament within a given length. The machinery of the *Lion* is being constructed by the Vickers Company, and will no doubt follow the lines so successfully evolved in preceding ships. Thus there will be four shafts, with turbines working in series on each side of the center line of the ship, each unit including a high-pressure and a low-pressure ahead turbine on separate shafts, and a high-pressure and low-pressure astern turbine respectively on each shaft. In accordance with recent practice, no cruising turbine will be adopted, but the high-pressure turbine is made of increased length, and several by-passes are fitted, so that the steam may enter at different stages according to the power to be developed. These and other improvements introduced will give a favorable economy even at low power. The results of the trials, alike from the point of view of efficiency and economy, are therefore certain to once more establish the satisfactory character of the design work produced by Sir Philip Watts and Sir Henry Oram, who are responsible for the ship and machinery respectively of all our warships.

It may be interesting to append the dimensions of typical cruisers of the last few years.

These data show the enormous step taken in length, displacement, speed, and power of armament. In this period of six years the proportion of length to beam has increased from

Name.	Length between perpendiculars.	Breadth.	Draught.	Displacement.	Speed.	Guns.
	feet.	*feet.*	*feet.*	*tons*	*knots*	
Devonshire (1904)....	450	68½	25	10,850	22.25	4 7.5-in., 6 6-in., 22 3-pounders.
Minotaur (1906).....	490	74½	26	14,600	23.00	4 9.2-in., 10 7.5-in., 16 3-in.
Invincible (1907).....	530	78½	26	17,250	26.00	8 12-in., 16 4-in.
Indefatigable (1909),	555	80	26	19,300	26.00	8 12-in., 16 4-in.
Lion (1910).....	700	88½	28	26,350	28.00	8 13.5-in., 16 4-in.

6.57 to 1 to 8 to 1. In view of this great increase in beam, the displacement of the cruisers per unit of length has greatly increased. Thus, in the case of the *Devonshire* class, it was equal to 24 tons per foot of length; in the *Minotaur*, to 29.8 tons per foot of length; in the *Invincible*, to 32.53 tons per foot of length; whereas, in the case of the *Lion*, the displacement is nearly 40 tons per foot of length.—"The Engineer."

H.M.S. Orion and Neptune.—The battleship *Orion* was launched at Portsmouth August 20th, and is the fifth vessel of the *Dreadnought* or improved *Dreadnought* type which has been launched from that dockyard.

The first keel plate of the *Orion* was laid on November 29th last year, so that only 264 days, including Sundays, elapsed between that time and the date of her launch.

As in the case of all warships built in recent years the Admiralty has exercised great secrecy regarding the design and structural details of this the latest of our battleships, and anything written concerning her must be regarded as being largely surmise. From what we can gather, however, it would appear that she is to be a ten-gun ship with five barbettes arranged on the center line, the second and fourth barbettes being raised so that they can fire ahead or astern respectively over the first and fifth barbettes. It is now practically certain that the guns will be 13.5 inches. The whole of the ten guns

will be able to fire on the broadside. It will also be seen that four guns can fire right ahead and four astern, while by firing slightly at an angle six guns can be made to bear very nearly straight ahead.

In the case of the *Neptune,* which immediately preceded the construction of the *Orion* at Portsmouth, and the displacement of which was very nearly 20,000 tons, the arrangement of the guns was quite different. There was one two-gun barbette on the center line forward, then a two-gun barbette placed on the port side, and further aft still another on the starboard side, and two two-gun barbettes right aft. By means of this arrangement six guns could fire dead ahead and eight dead astern, but though all ten guns could fire on the broadside, the angle of fire of one of the guns was limited. The secondary armament of the *Orion* is, we believe, to consist of twenty 4-inch guns, and her armor will vary in thickness from 4 inches to 12 inches.

The same uncertainty exists regarding the actual dimensions of the *Orion* as there is concerning her armament and armor. It is believed, however, that the vessel is 545 feet long between perpendiculars and 584 feet long overall, and that she has a beam of 87 feet. Her displacement is given variously between 22,500 tons and 23,500 tons. Perhaps if 23,000 tons be taken it will not be far from the mark. It is said that her full coal capacity will be 2,700 tons and her normal capacity 900 tons, and her designed speed is stated to be 21 knots, which will be obtained by four sets of turbine engines of an aggregate horsepower of 27,000, working on four shafts, there being eighteen Babcock & Wilcox boilers, which can, if required, burn oil fuel, of which about 1,000 tons is to be carried.—" Engineering."

New Protected Cruisers.—It is understood that private builders have sent in tenders to the Admiralty for two protected cruisers of the *Bristol* type. They are two of five to be laid down, the other three having been allocated, two to Pembroke and one to Chatham. Tenders for machinery for

armored ships now building are also being forwarded. It is stated that five Clyde firms have tendered for the cruisers.—"Page's Weekly."

Wreck of the Bedford.—Officers of our service, who were attached to vessels of the Atlantic Fleet in 1905, and enjoyed the hospitality of her officers, will note with particular regret the loss of the armored cruiser *Bedford,* with eighteen men of her engineer's force, on August 21st. The *Bedford* formed a part of the squadron under Admiral Prince Louis of Battenberg, which visited the United States during that year. She was wrecked on the southwest end of Quelpart Island while on passage from Wei-hai-wei to Nagasaki.

Steam Trials of H.M.S. Gloucester.—H.M.S. *Gloucester,* the fourth of the second-class protected cruisers ordered under the program of 1908–9, has just completed her trials, and Messrs. William Beardmore & Co., who built and engined the vessel at their Naval Construction Works, Dalmuir, are to be congratulated upon the results, alike as regards the speed and machinery efficiency. The speed attained on the full-power trial was the best so far recorded for any ship of the class—namely, 26.296 knots, when the turbines were developing a collective power of 24.335 shaft horsepower, while the consumption of fuel was 1.14 pounds of coal, and 0.42 pounds of oil. The vessels were designed to maintain a speed of 25 knots, and the trials of the four so far completed—the others being the *Glasgow, Liverpool* and *Newcastle,* built respectively at Fairfield, Barrow and Elswick—show that 26 knots can easily be realized. This is particularly satisfactory in view of the dimensions of the vessels, which are 430 feet long between perpendiculars and 47 feet beam. The displacement is 4,800 tons on a mean load draught of 15 feet 3 inches. They are intended for scout duty, and are fitted with two 6-inch guns and ten 4-inch guns. The four vessels so far tried have Parsons turbines mounted on four shafts, each with an ahead and an astern turbine.

The *Gloucester* first carried out a progressive trial of over twelve hours divided into three equal periods, while the other trials included a twenty-two hours' run and two eight hours' runs. The speeds attained in each of this complete series of progressive runs, with the power recorded, are as follows:

	S.H.P.	Knots.
4 hours' trial.....	2,053	12.837
4 hours' trial.....	5,513	17.55
4 hours' trial.....	9,400	20.8
22 hours' trial.....	13,968	23.447
8 hours' trial.....	18,983	25.084
8 hours' trial.....	24,335	26.296

The twenty-two hours' trial at 13,900 and the eight hours' trial at 18,000 shaft horsepower were continuous, and constituted the fuel-endurance tests. It was found that at 13,968 horsepower the fuel consumption per shaft horsepower per hour was only 1.59 pounds, and at 18,983 shaft horsepower it was 1.48 pounds; as already shown, very satisfactory results were also got at full power. These four vessels complete the cruisers of corresponding type given out eighteen months ago; the fifth—the *Bristol*—has been fitted with Brown-Curtis turbines by Messrs. John Brown & Co., Limited, of Clydebank.

H.M. Protected Cruiser Newcastle.—The second-class protected cruiser *Newcastle* is typical of the "Town" class, five of which were ordered under the 1908–9 program, and four under the 1909–10 program, while provision is made in this year's Estimates for the laying down at an early date of three more. These vessels are light unarmored ships, intended for reconnaissance work in association with the more powerful armored cruisers of a fleet, and, being possessed of a speed of 26 knots, they will prove most serviceable. They are not intended to penetrate the screen of the enemy's force, and thus the armament is limited to 6-inch bow and stern chasers, with five 4-inch guns on each broadside. They also carry two Maxim guns, and have two submerged tubes for 18-inch tor-

pedoes. They may be regarded as developments of the *Attentive* scout class, which, although of the same legend speed—25 knots—only attained this result under favorable conditions, as they were fitted with reciprocating engines, whereas the speed of the four "Town" cruisers so far tried ranges between 25.8 and 26.296 knots—a difference largely due to power developed, which varied between 22,500 and 24,335 shaft horsepower in the respective ships. But as the machinery in all of these ships is of practically the same design, each may be credited with a 26-knot speed. In addition to this speed advantage, the new cruisers, being some 60 feet longer and of nearly 900 tons greater displacement than the vessels of the *Attentive* type, can keep the sea more easily; they have a greater armament, since the *Attentive* class had only ten 3-inch quick-firers and eight smaller guns, while the fuel supply carried gives them a much wider radius of action.

The *Newcastle,* like the other ships of the class, has a length, between perpendiculars, of 430 feet, a breadth, extreme, of 47 feet, and at a mean draught of 15 feet 3 inches, the displacement is 4,800 tons. The lines are exceptionally fine, being a development of those of the *Attentive* class. A noticeable feature is the outward large curve of the bow wave, due to the flare out above the load line. This will make the vessels dry boats in heavy weather, a great desideratum for a scout, which may have to undertake a running fight after an enemy's destroyer or scout. The high forecastle is also contributive to the same result. The funnels, which were considered rather short, have been lengthened.

An important feature of the machinery of this class of cruiser is that in the case of two of the ships—the *Bristol* and *Yarmouth*—Brown-Curtis turbines are being fitted by Messrs. John Brown & Co., Limited, Clydebank, so that the comparison may be made with the performance of the Parsons turbines in the seven other ships now being built. These latter so far have given splendid results, the water consumption per shaft horsepower per hour for all purposes having in at least one case been 12.60 pounds at full power and 13.16 pounds at half

power, while the fuel consumption for all purposes has been 1.6 pounds at full power and at half power.

The general arrangement of the Parsons turbine installation follows recent practice, there being four lines of shafting arranged in three separate engine rooms. The high-pressure ahead and separate high-pressure astern turbines are arranged on one shaft in each wing compartment. One low-pressure ahead turbine (in which is embodied the low-pressure astern turbine) is mounted on each of the inside shafts, being placed in a center compartment, which also contains the maneuvering gear for all the turbines. But in these cruisers, as in all warships subsequently laid down, there are no separate cruising turbines, a special series of blades for cruising purposes being fitted in each high-pressure turbine. Both turbines are within the one casing, and the large eduction pipe is a prominent feature.

The condensers in these cruisers are located abaft the turbines, the port and starboard condensing plant being in separate watertight compartments. The condenser and air-pump installation on the *Newcastle* is notable, as the former is of the "Uniflux" type, and the latter of the "Dual" type, both introduced by Messrs. G. & J. Weir, Limited, Cathcart. The gain due to high vacua on the steam turbine is a point which emerged early in its development, but it is only recently that the majority of engineers have appreciated the extent of this gain. From 26-inch to 27-inch vacuum there is a gain of about 4 per cent.; a further gain of 5 per cent. if the vacuum is increased to 28 inches; and a still further gain of 6 per cent. to 7 per cent. when it is increased to 29 inches. The opportunity afforded of thus adding to turbine efficiency was early seized by Messrs. Weir, who had been identified with the introduction of independent air pumps for naval vessels, and consequently had a considerable experience in condenser conditions. The results of these air-pump installations caused them to direct their attention to the design of the main condenser as a chief factor in the attainment of high vacua, and after a long series of trials and experiments, they ultimately

produced the design of condenser now known as the "Weir-Uniflux." The two "Weir-Uniflux" condensers for the *Newcastle* were built by the Wallsend Slipway and Engineering Company, Limited, by arrangement with, and to the designs of, Messrs. Weir. The contour is of the pear or heart-shape characteristic of the "Uniflux" design, to secure a uniform velocity of steam-flow over all the tubes, and the cooling surface is considerably less than that hitherto considered necessary for ordinary condensers for equivalent duties. The shells are of steel, with brass tubes and tube-plates and gun-metal doors. For the designed power of 22,000 shaft horsepower of the *Newcastle's* turbines the cooling surface, with condensers of the usual cylindrical design, would have been about 22,000 square feet. The "Uniflux" condensers fitted to the vessel, however, have a total cooling surface of 16,820 square feet, representing a saving in weight and space, which is of vital value in a crowded naval engine room. The following figures show the condensing plant performance at full power—22,000 shaft horsepower:

Sea temperature, degrees F.....	46
Vacuum at top of condenser (corrected to 30 inches barometer), inches.....	29.02
Circulating discharge temperature, degrees F..	74
Hot-well temperature, degrees F.....	74
Temperature equivalent to vacuum, degrees F..	79

In conjunction with the condensers Messrs. Weir have also furnished four of their "Dual" air pumps 22 inches by 12 inches by 16 inches. These pumps are of the twin type, but have the distinctive feature of combining a wet and a separate dry pump in one apparatus. Whenever the contents of the condenser are handled by one pump, or its equivalent—a twin pump with common suction—the hot-well temperature for a given vacuum is dependent on the amount of air leakage and air-pump capacity. By the use of separate pumps handling the air and water, the dry pump having a cold injection non-returnable to the feed, this phenomenon can be largely overcome. In this

"Dual" pump the dry-pump portion can be kept at a comparatively low temperature, as a closed supply of injection water continuously cooled circulates through the pump, densifying its contents and consequently increasing its efficiency. On the other hand, the wet-pump portion handles the water from the condenser at approximately the steam temperature, and thus secures a high hot-well temperature, as, of course, none of the feed need be cooled down to secure the high vacuum which is obtained by the dry pump.

The boilers, twelve in number, of the small-tube type, are arranged in three separate boiler rooms, and in each boiler room there will be one main and one auxiliary feed pump. The boiler rooms are adapted for use with the closed-stokehold system of forced draft, the supply of air being provided by steam-driven fans. Oil pumps are provided for both oil fuel and forced lubrication purposes.

The *Newcastle* has completed all her trials, and will be commissioned at an early date. The 30 hours' endurance trial was divided into two parts; the vessel was required to run for eight hours when developing about 18,000 shaft horsepower. With a mean air pressure in the stokehold of 1.22 inches and an average for all four propellers of 461.6 revolutions per minute, the shaft horsepower averaged 18,742, while the coal consumption was 1.65 pounds. The vacuum was 29.02 inches. The subsequent 22 hours of this run were to be at 13,500 shaft horsepower, and at the close it was found that under easy steaming conditions 14,038 shaft horsepower was maintained at 420.5 revolutions, the coal consumption being 1.85 pounds, and the vacuum 28.76 inches. On the full-power trial of 8 hours' duration the power averaged 24,669 at 518.2 revolutions, the coal and oil consumption being respectively 1.036 pounds and 0.171 pounds per shaft horsepower.

At each power six runs were made over the measured mile at St. Abb's Head, and the results are tabulated:

Mean revolutions.....	422.1	465.8	522.9
Shaft horsepower.....	14,051	19,116	25,417
Speed, knots.....	23,342	24,841	26,266

These results are most creditable, not only to the technical officers at the Admiralty, who are responsible for the design of the ship and machinery, but also to Sir W. G. Armstrong, Whitworth & Co., Limited, who built the ship, to the Wallsend Slipway and Engineering Company, Limited, who constructed the machinery, and to Mr. Andrew Laing, their managing director.—" Engineering."

The Brown-Curtis Turbine Installation in H.M.S. Bristol. —The trials of His Majesty's second-class cruiser *Bristol,* completed on Tuesday last, mark a distinct step in the application of turbines for ship propulsion, as the system of propelling machinery adopted in this vessel differs essentially from that hitherto applied in British warships, and each success achieved contributes towards progress in the ceaseless and unending pursuit of efficiency. The application of the steam turbine to the propulsion of high-speed ships, particularly warships, has conferred great tactical advantages, apart altogether from the practicability of attaining speeds greatly in excess of those possible with piston engines, and apart also from the economy realized; and the great service the Hon. C. A. Parsons has rendered to the Royal and merchant navies is therefore universally acknowledged. In the future, as in the past, his genius and research work will evolve developments, and these, it is reasonable to assume, will be assisted by the work of others towards the same end—the advancement of the turbine system. The *Bristol* is the first British ship tried with a type of turbine other than that designed by Mr. Parsons; and the fact that so long a time has elapsed before such a trial was made, combined with the extent to which the Parsons system has been preferred in all maritime countries, affords eloquent testimony of the efficiency of that system. At the same time it justifies a lengthy review of the performance of the *Bristol* on her trials, and of the light which the results throw on the possibilities of the advancement of the turbine system as applied to marine purposes.

Attention may first be directed to the progress of turbine

propulsion. After a long series of experiments with land turbines, unexcelled in the history of mechanical engineering for their ingenuity and resourcefulness, Mr. Parsons produced his first commercially successful marine turbine in 1894 for the *Turbinia,* devoting research work with equal assiduity to the propeller problem. In 1899 the Admiralty first ordered an installation of turbine machinery of the Parsons type for the propulsion of a warship, and the results then got corroborated the marvellous performance achieved in the *Turbinia,* which was built for Mr. Parsons' company. The British Admiralty, satisfied with the result, moved step by step, first in destroyers and next to the third-class cruiser *Amethyst,* built in 1905. The high economy attained by this vessel gave confidence to the technical officers, and afforded them full justification in recommending the adoption of the system in the battleship *Dreadnought.* Since then no British warship has been fitted with any other form of machinery than Parsons turbines.

Foreign Governments have followed in the same lines, the Parsons turbine, by reason of increasing success realized as a consequence of fuller experience, proving universally acceptable, alike from the standpoints of reliability and economy. It was only natural, however, that inventors of other turbines suitable for land work should endeavor to follow in the footsteps of the pioneer in adapting their types for marine propulsion, and equally natural that in other countries consideration should be given to the systems invented in such countries. Thus in America the system invented by Mr. C. G. Curtis secured favor along with the Parsons turbine. In France the Rateau system has also been considered, although not extensively applied. In Germany the Zoelly system has been tried to some extent. Patriotic considerations had something to do with the trials of alternatives to the Parsons turbine, but nevertheless the latter is adopted in the great majority of warships throughout the world, as well as in high-speed merchant ships. Japan fitted the Curtis system to two battleships, but the decision to adopt this particular design was no doubt partly due to the fact that the hulls had been far advanced on the under-

standing that twin-screw reciprocating engines were to be fitted, and that when a change was decided upon in favor of turbines the Curtis arrangement was adopted because it was more adaptable under the particular circumstances named. In Germany, too, competitive trials were instituted, not only between the Parsons design and the Curtis system, as manufactured by the A.E.G., but also with other designs of turbines.

The British Admiralty, always open to consider improvements which may conduce to advancement in efficiency, also decided to install other systems than that conceived with such genius, and developed with such painstaking care and extensive research work by Mr. Parsons. Consequently, when Messrs. John Brown & Co., Limited, took up the manufacture of the Curtis turbine and submitted designs and proposals of this system in connection with a cruiser of what is now known as the "Town" class, the Admiralty were favorable to entertain seriously the trial of the system in such a vessel, and the *Bristol* is the result.

The reasons which led the well-known Clydebank firm to take up the construction of this impulse type of turbine were "(1) the potential advantages of acquiring experience with a type of marine turbine capable of using superheated steam, as in land installations; (2) the attainment of economy at low powers, without the disadvantage of very close-fitting parts, and the extremely fine adjustments entailed thereby; and (3) the simplification of the connections and general engine-room arrangement, and also the expectation of attaining higher propeller efficiency by an increase in size of the individual propellers."

The potential advantage of using superheated steam cannot be gainsaid. Highly economical results have been got on land stations, notably in connection with the generation of electricity by Parsons and other turbines, as well as piston engines. In connection with marine installations, however, there is a difference of opinion, not on the question of the economy attainable, but rather as to the reliability of superheaters, due to the possibility of salt water getting into the

boilers and thence to the superheaters through leaky condensers, with consequent corrosion and pitting in the superheater tubes.

Superheated steam, it should be recalled, is no novelty in the naval service, a fact which was dwelt upon in the lucid address which the Engineer-in-Chief of the Navy—Engineer Vice-Admiral Sir Henry Oram, K. C. B.—delivered in November last, as President of the Junior Institution of Engineers. In this he pointed out that from 1863 to 1870 "Admiralty boilers were always specified to have superheaters, but they had thus only a brief career, and were abandoned in new ships at the later date, and did not reappear in the Navy" until the building of H.M.S. *Britannia* in 1904. This battleship, which had reciprocating engines of 18,000 indicated horsepower, was fitted with twenty-one Babcock & Wilcox boilers, six of which had superheaters incorporated in their design, with the result that, at cruising speed, when these boilers only were in use, the water consumption was 13½ per cent. lower than when a corresponding number of boilers without superheaters were in use. The actual amount of superheat was 93 degrees F., with a gauge pressure of 200 pounds. Careful observation was, of course, taken of the endurance of the boilers with superheaters, and, "at the end of 3½ years, although pitting was found to be taking place, the tubes were considered good for several years' further service." This fact was no doubt regarded as justification for the adoption of the system of superheating proposed to be installed in the *Bristol*.

Preparatory to putting proposals before the Admiralty for the installation of Curtis turbines in a warship, Messrs. John Brown & Co. decided to construct a complete installation for experimental purposes, in order to determine important questions of detail in design. This procedure was further necessary because the Admiralty are naturally opposed to the adoption in warships of any appliance which is more or less experimental, unless there has first been a complete demonstration of efficiency. The tests therefore carried out at Clydebank were not only to improve the design of the Curtis turbine, but

to establish the fact that in agreeing to the adoption of such a system in a warship the Admiralty might have good grounds for anticipating that the results in the ship would be comparable with the marvellously successful attainments of the Parsons turbine. All the tests made at Clydebank with the experimental turbine were therefore carried out in the presence of Admiralty officials, and many improvements or safeguards which they found desirable as the result of the experience gained in these tests were incorporated in the design of the machinery for H.M.S. *Bristol*. This prudent procedure adopted has been justified by the results which we are now able to record in connection with the official trials of the *Bristol* in which the same mileage per unit of water consumption has been obtained as in the four ships of the same class which have preceded her, and which are fitted with Parsons turbines, embodying the inventor's latest improvements at the date of their construction.

Before describing the *Bristol*, we may deal with the experimental researches carried out at Clydebank, as these had such a direct bearing on the evolution of the design of the *Bristol's* machinery. It was recognized that the compactness, and the resulting gain in simplicity of engine-room arrangement and in the space occupied, which was claimed as the great advantage for the impulse type of turbine, had been partly obtained at the cost of economy—a fact apparent from the trial results of the only two Curtis turbined ships then running—viz: the Southern Pacific Company's steamer *Creole* and the Hamburg-Amerika Company's excursion steamer *Kaiser*. The Curtis turbine, as fitted in the *Creole*, of 8,000 total shaft horsepower, had only seven wheel stages for driving the ship ahead, and two wheel stages for going astern. The turbines in the United States cruiser *Salem*, which were designed to give 16,000 shaft horsepower, but developed 18,000 shaft horsepower, are practically duplicates of those fitted in the *Creole*, differing only in the increased area through the several nozzles to suit the greater quantity of steam to be passed. The builders of these turbines thus demonstrated the great

flexibility of the Curtis system, in allowing turbines of so different a power to be made from the same patterns, but they seem to have neglected the vital point of economy, and it is difficult to understand at the present day how it could have been expected that the *Salem's* machines with only twenty-two rows of moving blades in the ahead turbine would be comparable in economy when running at high powers with the already fully developed Parsons marine turbines.

The technical staff at Clydebank were confronted with the problem, solvable only by experiment, of finding from actual turbine tests the relation between the calculated and actual efficiencies, so that the precise number of stages and requisite blade velocities—and no more—were provided to ensure that the desired water consumption would not be exceeded, and that at the same time the advantages of compactness and simplicity would not be minimized. This had to be determined for both the ahead and astern turbines, and at the experimental station this research work was carried out, and the design of the turbines of the *Bristol* evolved.

The experimental plant had to be erected in a shed close to the banks of the River Clyde, to obtain a ready supply of circulating water. The two boilers were of the Babcock & Wilcox marine type, fitted with superheaters. Arrangements were made for disconnecting the superheater, so that comparative tests of the turbine could be made with saturated steam.

The turbine shaft was prolonged to work through a water brake, giving a constant resistance, although not sufficiently precise in construction to measure satisfactorily the power. Between the water brake and the turbine, therefore, a torque shaft was interposed, and on this was fitted a Hopkinson-Thring torsionmeter, which was officially calibrated, and was checked before and after each day's trial. The condenser was of the Weir "Uniflux" type.

The first Curtis turbine designed and built for experiment by Messrs. John Brown & Co. was precisely on the lines of the *Creole*, except that the number of wheel stages was increased from seven to eleven in order to improve the economy. This

turbine had thirty-four rows of moving blades in the ahead turbine, and illustrated very well the compactness and ample blade clearance and other general features of this type.

In ships fitted with this wheel-stage type of turbine practically the entire thrust of the propeller is taken on the thrust block, as in reciprocating-engined ships. While the first experimental turbine was being built, it was therefore decided to make the necessary new parts for the second arrangement of turbine, in which a drum replaced the last four wheel stages, so that by utilizing the steam pressure on the end of this drum the thrust of the propeller could be balanced. Incidentally also the substitution of this drum for the later wheel stages enabled a greater number of rows of blades to be obtained with the same over-all length of turbine, eighteen as compared with twelve. The total number of blade rows was thus increased to forty—a number required at the designed blade velocity to keep the water consumption per shaft horsepower below 13 pounds, with a superheat of about 50 degrees F.

For further experiment and comparison, a second drum carrying blades of the reaction type was built, and the second drum, a new section of casing, together with the same early wheel stages, were combined to give this third experiment. Comparative tests were made between the all-impulse and the impulse-reaction types; but the difference in the results was inappreciable, and the all-impulse design was accepted, with the requisite increase in the number of rows of blades, to be referred to later.

The same thorough course of development was carried out with the astern turbine wheels, the first arrangement having two-wheel stages, very similar to those fitted in the United States cruiser *Salem*. The results of the experiment on these two-stage arrangements showed a very poor efficiency; consequently a very short impulse drum was added, with a resultant improvement. In order, however, to meet the Navy's requirements of developing with the astern turbines a power equal to half the ahead power of the main turbines, it was requisite that the steam consumption should be brought down

to about 25 pounds of steam per shaft horsepower, and, to attain this, it was found necessary to still further lengthen this additional drum.

Figure 1 gives a curve of the final results obtained as a mean of several repeated runs, in the presence of Admiralty officials. These results apply to the second experimental turbine. It will be seen that at one-fifth power the steam consumption with a very moderate superheat was only 17.9 pounds per shaft horsepower per hour. At half power this figure was reduced to 14.2 pounds, the degree of superheat being still very moderate. At 75 per cent. of the total power the consumption was only 13.4 pounds, whilst at full power, with the slightly greater, but still low, superheat of 50 degrees F., the steam consumption was 12.9 pounds.

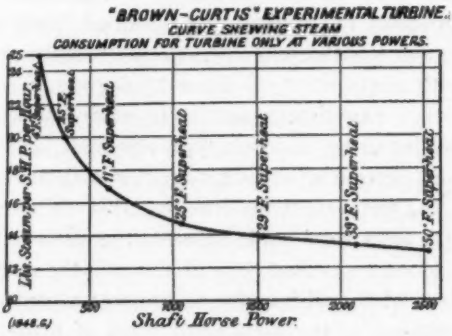


Fig. 1.

Fig. 2 is interesting, as it shows the gain obtained from the use of superheated steam as compared with the consumption with saturated steam. The results conformed well with the well-known assumption that for every 10 degrees of superheat 1 per cent. gain in economy is obtained. It will be seen, however, that the line is slightly curved, the gain being rather more for the few degrees of superheat than for the higher rate. This is in full conformity with the experiments by Knoblauch and Jakob, which showed that near saturation the specific heat

of superheated steam was considerably more than at still higher temperatures.

The results obtained in the final astern trials, with a superheat of 50 degrees F., are as follows: With full boiler power, 1,403 shaft horsepower with 23.05 pounds water consumption; with half boiler power, 577 shaft horsepower with 27.45 pounds water consumption per shaft horsepower per hour. The actual results attained on the trials of the *Bristol* were equally favorable. With half boiler power and with the turbines developing 4,390 shaft horsepower, the water consumption for the turbines and all the auxiliary machinery was 35.5 pounds per shaft horsepower per hour, the degree of superheat being 40 degrees F. With all the boilers in use, the shaft horsepower was 12,450, with a steam consumption for all purposes of 28.2 pounds per shaft horsepower per hour, and for the main turbines only 23.9 pounds per shaft horsepower per hour, the degree of superheat being again 40 degrees F.

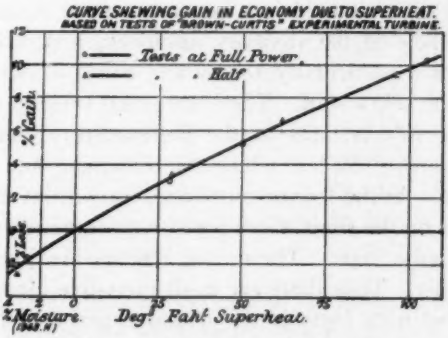


Fig. 2.

As regards details in design which were evolved during the experiments, special note may be made as to the turbine-spindle glands. It was found that the original stuffing box could be materially simplified on the lines of the design of the British Thomson-Houston Company, of Rugby. The packing consists of five carbon bushes placed in separate grooves and held

together by springs, a leak-off or steam supply being provided between the second and third rings.

And now we may turn to the *Bristol* and the results of the official sea trials. The *Bristol,* like other ships of the class, has a length between perpendiculars of 430 feet, a breadth of 47 feet, and, at a mean draught of 15 feet 3 inches, the displacement tonnage is 4,800. The boilers, twelve in number, of the small-tube Yarrow type, are arranged in three separate boiler rooms. Each boiler is fitted with a very simple arrangement of superheater. The total grate surface is 830 square feet, the total generator heating surface 45,366 square feet, and the total superheater heating surface 3,720 square feet. At full power the boilers burn oil as well as coal, but on all the other trials only coal was allowed, and no difficulty whatever was experienced at any time in obtaining an ample amount of dry steam.

The *Bristol's* machinery consists of two Brown-Curtis turbine units, each self-contained and entirely independent of the other, and each situated in a separate engine room, together with the whole of the auxiliary machinery appertaining to it. This makes a particularly roomy and easily-worked arrangement for the ship's staff. There are seven velocity-compounded impulse wheels; that at the high-pressure end has four rows of moving blades, while the remaining six have each three rows. At the low-pressure end, on a drum mounted on wheel discs on the shaft, there are twenty-three rows of blades of the impulse type. These, of course, are pressure-compounded only. Thus there are in all forty-five rows of blades, as compared with forty in the second experimental engine of the same type. The blades are of the usual cold-drawn brass. The astern turbine is constituted by two impulse wheels, each with five rows of moving wheels, followed by an impulse pressure-compound drum, with seven rows of blades.

The two condensers are each placed in the wings in a separate watertight compartment, in which there is no other machinery whatever; so that even if these compartments were pierced and flooded no difference would be made to the work-

ing of the engines. The condensers are of the well-known Weir "Uniflux" type, which have been fitted in the whole of these vessels.

The program of official trials to be carried through was precisely the same as those of the other ships: a 30 hours' endurance trial—22 hours at 13,500 shaft horsepower, and 8 hours at 18,000 shaft horsepower—and a full-power eight-hours' trial, the designed power being 22,000 shaft horsepower, for which the legend speed was 25 knots. There were also trials at progressive powers, astern, and other maneuvering tests.

First, with regard to the speed attained, the following gives for the various trials the mean power on the trial and the mean speed attained on the runs on the measured mile made during each trial:

TABLE I.—*Powers and Speeds of H.M.S. Bristol on Official Trials.*

	Mean power throughout trial. *S.H.P.*	Mean speed on measured miles. *Knots.*
4 hours' trial	1,839	12.96
4 hours' trial	4,910	17.66
4 hours' trial	9,417	21.17
8 hours' trial	19,130	25.17
22 hours' trial	14,300	24.06
8 hours' trial, full power	24,227	26.84

Table II gives the speed on each of the six runs made on the measured mile on the eight-hours' full-power trial.

The true mean speed, determined by the Admiralty formula, is thus 26.84 knots, the mean power on the eight-hours' trial being 24,227 shaft horsepower.

The speed attained on the full-power trials of the ships of the class have varied but slightly, and generally in conformity with the power developed, although differences in the design of propellers, and possibly in weather conditions, have had something to do with these variations. The average of the

mean power on the eight-hours' trial for the four preceding ships was 23,800 shaft horsepower, and the average of the true mean speeds attained on the measured mile 26.13 knots, speeds ranging from 25.8 knots to 26.296 knots. In the case of the *Bristol* the power—also on the eight-hours' trial—was 24,227 shaft horsepower, and the speed on the measured-mile runs 26.84 knots.

TABLE II.—*Speed on Six Measured-Mile Runs on Full-Power Trial, September 27, 1910.*

	Knots.	
Run No. 1	26.47	26.35
2	26.239	
3	27.191	26.92
4	26.647	
5	27.439	27.04
6	26.647	

On the trial at 18,500 shaft horsepower again, the average of the mean powers on the eight-hours' trial of the four preceding ships was 18,650 shaft horsepower, and the average of the true mean speeds on the measured-mile runs, 24.98 knots. The power on the corresponding trial of the *Bristol* was 19,130 shaft horsepower, and the speed 25.17 knots.

On the twenty-two-hours' trial the *Bristol's* mean power was 14,300 shaft horsepower and the measured-mile speed 24.06 knots, while the average of the mean powers of the four ships on the whole run was 14,016 shaft horsepower, and the average of their measured-mile speeds 23.56 knots.

Water consumption results are plotted on the diagram, Fig. 3. The most noteworthy point is the very large saving in the total steam consumption ensured by turning the exhaust from the auxiliary engines into the low-pressure end of the turbine. The practical effect of this is that for a large range in power the steam consumption of the auxiliaries is reduced by half, so far as the demand on the boilers is concerned. The diagram is further interesting because it shows the large steam consumption of the auxiliary machinery. Running with open exhaust

the auxiliaries take rather more than one-fifth of the total steam supplied for all purposes at full power. There is thus once more emphasized the great importance of working with closed exhaust, which discounts to a considerable degree the thermodynamical inefficiency of auxiliary machinery. It will be seen also that the increase in consumption between full power and half power is moderate, the curve being very flat. Taking the results for the turbines only, the steam consumption at one-fourth full power is about 16 pounds per shaft horsepower per hour, which a few years ago would have been considered a very favorable figure for reciprocating engines even at three-quarter load. At three-quarter power the consumption of the turbines was 12.7 pounds, and at full power 12.2 pounds. The significance of this last figure is established by reference to the improvement in the efficiency of the turbine machinery in recent years.

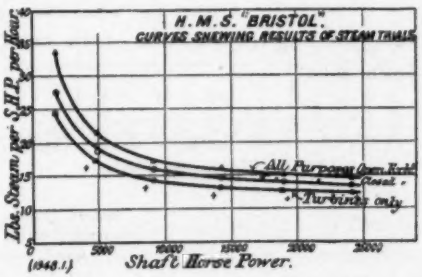


Fig. 3.

In 1907 a consumption of about 13½ pounds of steam per shaft horsepower for the main turbines only was considered most satisfactory for battleship machinery. The turbines of subsequent ships were, however, gradually improved after experience, with the result that, in Sir Henry Oram's paper already referred to, it is stated that the average steam consumption of the later battleships has been very slightly over 13 pounds per shaft horsepower for the main turbines. In the still larger turbines, of 41,000 horsepower, of the *Indomitable* class, the remarkable economy of slightly over 12 pounds of

steam for the main turbines was recorded. It was, therefore, decided to design the turbines for the *Bristol* with sufficient expansions to ensure the steam consumption not exceeding 12½ pounds for the main turbines alone at full power—in the expectation that the higher propeller efficiency resulting from the relatively large twin-screw propellers installed should give a combined propulsive result equal to what might be expected in her sister ships fitted with Parsons turbines. That these results have been attained on actual trial is remarkable evidence of the correctness of the calculations and the reliability of the data on which they were based. The actual steam consumption was 2½ per cent. less than that for which the turbines were designed, an approximation to the anticipated result, which establishes the value of the research work done, and the accuracy of the scientific methods adopted.

In Table III there is given the steam consumption for the turbines of the *Bristol* at various powers, and alongside the figures the corresponding trials of one of the sister ships fitted with Parsons machinery.

TABLE III.—*Water Consumption per Shaft Horsepower Hour (Main Turbines Only.)*

	Brown-Curtis turbined ship. *pounds.*	Parsons turbined ship. *pounds.*
18,000 shaft horsepower	12.65	12.1
Full power	12.2	11.75
8,800 shaft horsepower	14.1	13.6
4,400 shaft horsepower	17.8	15.8
1,600 shaft horsepower	24.6	24.7

The trend of the steam curve would be very similar if plotted for both ships, the Parsons turbine, however, showing the higher efficiency. This difference is discounted by the higher efficiency of the propellers of the *Bristol*, rendered possible, no doubt, by the adoption of the two-shaft system instead of the four-shaft arrangement in the other ships, and the conse-

quent better disposition of the screws relative to the hull of the ship. Fig. 4 shows the miles run per ton of steam consumption for all purposes. This measure of efficiency appeals more to the tactician, as it determines the radius of action, the water evaporated per ton of coal being a fairly constant unit. The radius of practically all the ships, apart altogether from the type of turbine used, is about the same as in the *Bristol*. At full speed—0.122 mile per ton of water evaporated; at three-quarter power (giving the continuous sea steaming speed) 0.25 mile; and at 13 knots, the usual cruising rate, 0.50 mile. While the performance of the *Bristol* and her ma-

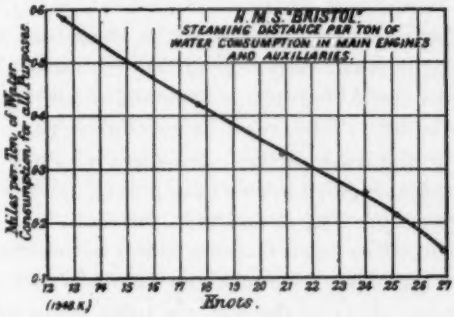

H.M.S. "BRISTOL".
STEAMING DISTANCE PER TON OF WATER CONSUMPTION IN MAIN ENGINES AND AUXILIARIES.
Miles per Ton of Water Consumption for all Purposes
Knots.
(1948 K.)

Fig. 4.

chinery, viewed from every point of view, is thus highly favorable, it is reasonable to assume, especially as this is the first Brown-Curtis installation, that further developments will be made and still higher efficiency be achieved with the experience gained, as has been the case in successive Parsons turbine installations. As it is, the *Bristol*, like all the other ships of the class, has proved exceedingly satisfactory alike as regards efficiency of machinery and total propulsive efficiency, the lines being specially designed for high speed. On this point Sir Philip Watts, K. C. B., the Director of Construction at the Admiralty, is to be congratulated, while engineers will greatly appreciate this further proof of that policy of advancement which so characterizes Sir Henry Oram, K. C. B., the Engineer-in-Chief of the Fleet.—" Engineering."

The Brazilian destroyer Sergipe left Messrs. Yarrow's yard at Glasgow October 24th, to start on her voyage to Brazil. The *Parana* left in the preceding week. These two vessels are the last of the ten destroyers ordered by the Brazilian Government from Messrs. Yarrow, and their departure witnesses the completion of a most important and efficient addition to the Brazilian Fleet. It is interesting to note that in spite of the firm's move from London to Glasgow during the early part of the execution of this contract, the vessels have been delivered some months before the contract date.

Yarrow Boilers for British Warships.—It is not long since that the idea was entertained that the small-tube boiler was only suited for small ships, on the somewhat simple theory that big ships needed big-tube boilers, and small ships required small-tube boilers. The remarkable results got with these boilers led to the trials of them in a modified form for larger ships, and these modified boilers have proved most satisfactory. Where high air pressure is desirable, the small-tube boiler still holds the field, as so-called Express boilers in torpedo craft and the smaller high-speed cruisers, but similar types have been introduced into ships of the line, the tubes being made somewhat larger. Thus in destroyers the diameter of the tubes varies from 1 inch to 1½ inches, while in the larger ships the diameter is 1¾ inches, which is the standard size recommended by Messrs. Yarrow. Other modifications, of course, have been made, the most recent, by Messrs. Yarrow, being to pass the feed water through the outer tubes, where the water is heated before it enters the steam-generating tubes proper. At the same time any sediment or grease is deposited in these outer tubes, where it is least harmful, with the result that the tubes nearest the fire are not only more efficient, but have greater durability, in view of the absence of incrustation and deposit due to the saline matter and oil in the feed water. A return has recently been issued showing the number of Yarrow boilers fitted in warships since 1905, including those now being built, and it is a striking fact that the number of battleships

and cruisers so fitted is thirty-one, with a total horsepower of 804,285, whereas the number of battleships and cruisers fitted with other boilers is fifteen, with a total horsepower of 393,804. We have thus in four years built large ships with a total horsepower of practically 1,200,000, of which two-thirds is developed by steam from Yarrow boilers. Since 1907 fifty-eight torpedo-boat destroyers have been fitted with Yarrow boilers, the total horsepower being 795,500; this is a very large proportion of the total of seventy-three vessels, of 994,000 horsepower, laid down within three years.

Messrs. Yarrow & Co., of Glasgow, have just received an order for a high-speed motor boat for service at Buenos Aires. The vessel will be 60 feet long by 9 feet beam: propelled by five sets of internal-combustion engines of the Yarrow-Napier design, of approximately 300 total horsepower, designed to give her a speed of 25½ knots. She will be practically similar to the *Mercury II*, built by Messrs. Yarrow for the British Admiralty. It may be remembered that His Late Majesty King Edward and a Royal party went for a trip in the Solent in the *Mercury II* during Cowes Week, 1906.

Australia.—Two destroyers, the first of the new Navy of the Australian Commonwealth, sailed in September from England after completion of their acceptance trials. These vessels represent the latest practice in torpedo-boat destroyers, and have a length between perpendiculars of 245 feet, a breadth molded of 24 feet 3 inches, and a depth of 14 feet 9 inches, their displacement being about 700 tons. The machinery is of the Parsons type, driving triple screws, with a high-pressure and two low-pressure turbines for steaming ahead, and corresponding arrangements for going astern. There are also high-pressure and intermediate cruising turbines. The three boilers are of the Yarrow type, using oil fuel. On the full-power trials the *Paramatta*, which was the first vessel tried, attained a mean speed for eight hours of 26.869 knots, when the turbines were developing 11,212 shaft horsepower, while the *Yarra*, on the same length of trial,

maintained a mean speed of 27.08 knots, with about the same shaft horsepower. The mean of the two fastest runs, with and against the tide, was 28.5 knots. In both cases the designed speed was 26 knots, so that the steaming performance was thoroughly satisfactory. Equally favorable were the results attained on the endurance trials. The contract required that the radius of action at 14 knots with a given quantity of oil fuel should be 2,500 nautical miles, but the consumption was so low as to enable the vessel to steam at this rate for nearly 3,000 nautical miles.

The armament includes one 4-inch quick-firing gun on the forecastle. One satisfactory feature of the 4-inch gun is that the gunner on each side (one for rotating and the other for elevating and laying the gun) is on a platform which rotates with the gun. There are three 3-inch 12-pounder quick-firing guns. These guns are on the upper decks, one on each side and one aft; they can all fire on either broadside. In addition the vessel carries on the upper deck three 18-inch torpedo-tubes.

Canada's Navy.—It is expected that the contracts for the construction of all the ships of the Canadian Naval Service will be awarded in October or November, and that the keels will be laid within a year from then. The intervening time will be required to get the building yards in readiness. The notice recently sent out to ship-building firms on the Admiralty list for tenders for the construction of four cruisers of an improved *Bristol* class and of six destroyers has elicited replies from three Canadian and two British firms. It is not improbable that there will be an alliance between certain Canadian and British interests in that connection. Canada has decided that all the new ships shall be built in the Dominion. Pending their completion the Dominion Government has purchased from the Imperial Government the cruisers *Niobe* and *Rainbow*. The *Rainbow* sailed for Esquimalt on the 20th August. She will be stationed in British Columbia waters and take part in the fisheries protection service. The *Niobe*, after un-

dergoing some alterations, will probably sail to the Gulf of St. Lawrence, where she will be used for training purposes, with headquarters at Halifax. When the *Niobe* has served her period of usefulness as a training vessel on the Atlantic she will become the parent ship of the destroyers. The *Rainbow*, as already explained, is outside the ordinary naval program, being primarily intended for fishery service, but at the same time she will serve as a training vessel on the Pacific Coast, and no doubt later on will become parent ship to the destroyers which will subsequently be built on the Pacific. It is not intended at first to divide the destroyer force, but to keep it together on the Atlantic Coast. The cruiser squadron, however, will be divided between the Pacific and Atlantic seaboards. In the case of both the *Niobe* and the *Rainbow* the Imperial Government have afforded great facilities in lending officers and getting the crews together.—"The Engineer."

Canadian Naval School.—According to recent information there was to be opened in October, at Halifax, a Naval School for Cadets of the Canadian navy. The number of midshipmen will be 30. The cadets will remain two years, after which they will become midshipmen, and will then go to sea for two years. After these two years of service as midshipmen those who have demonstrated their fitness will be appointed sublieutenants. After such appointment it is a question for the future whether in order to give greater development to their practical instruction they will be assigned to vessels of the British navy.

TURKEY.

Our Constantinople correspondent informs us that the committee having in hand the reorganization of the Turkish fleet has decided to order four destroyers and buy two large merchant vessels. These last will be used in the transportation of pilgrims to Mecca, and on the conclusion of this service, will be turned over to the Navy for service as transports.

It is also announced that on account of the early entry into service of the Greek armored cruiser *Aberroff*, the Turkish Government desires to buy one or more battleships. Those which have been ordered in England will not be ready before at least two years. With this object in view, the Turkish Government inquired of the great naval powers if they were willing to dispose of any of their naval vessels to Turkey. Germany alone responded to this appeal, and offered to sell two battleships of the *Brandenburg* class. After long negotiations, the two Governments came to an understanding, the price for the two vessels being fixed at 22,500,000 francs. The *Weissenburg* and the *Kürfurst Friedrick Wilhelm*, which are the two vessels sold, have the following characteristics: Displacement, 10,060 tons; length, 108 meters; beam, 19½ meters; draught, 7.4 meters; armor, 400 millimeters; armament, six 280-millimeter guns, eight 110-millimeter, eight 88-millimeter; engines, 10,000 horsepower; speed, 17 knots.—"La Vie Maritime."

It is reported that the Ottoman Government are about to place an order for battleships with the Armstrong group. The only point which has not yet been settled is whether one or two ships shall be ordered. The same informant confirms the statement of the "Tanin" with regard to the decision of the Government to buy no more old warships, and adds that an order for fifteen or twenty merchant vessels will shortly be placed in England.—"Page's Weekly."

MERCHANT SHIPS.

The Georgian, Kentuckian and Honolulan.—These vessels were constructed recently by the Maryland Steel Company for the American-Hawaiian Line. They are each of 6,100 tons gross, and their general dimensions are: Length over all, 429 feet 2 inches; breadth, molded, 53½ feet; depth, molded to shelter deck, 39 feet. There are three complete steel decks, fore and aft peaks, and forward and after deep tanks. There are four tanks in the double bottoms arranged for carrying fuel oil or water ballast and two for carrying fresh water. Seven watertight bulkheads, all extending to the upper deck, and the after peak extending to the shelter deck, make these vessels practically non-sinkable. The *Georgian* and *Kentuckian* have accommodations for 24 passengers, and the *Honolulan* for 48, which together with the officers' and crew's quarters are in steel deck houses above the shelter deck and in the after upper 'tween decks. A refrigerator room is located on the upper deck immediately aft of the engine house. The dining saloon is in the forward end of the shelter-deck house and is finished in quartered oak with paneled ceilings.

These vessels are capable of carrying 8,600 tons dead weight on 27 feet mean draught, and maintain a sea speed of 11 knots loaded. Special rooms are provided in the lower 'tween decks for bonded freight. The equipment for handling cargo is right up-to-date. There are three cargo ports on each side of the upper and shelter decks, also a small port in the coal bunker for carrying case goods while the ship is burning oil. The steam windlasses, winches, capstans, tillers and auxiliary tillers, with engine actuated by a telemotor from the steering house, pilot house and flying bridge were all furnished by the Hyde Windlass Company of Bath, Me. The passenger rooms are fitted with General Electric portable electric heaters.

The propelling machinery in each ship consists of a quadruple-expansion engine of the vertical, inverted, direct-acting, surface-condensing type, with cylinders 25, 36, 52 and 76 inches in diameter by 54 inches stroke, besides a complete equipment of auxiliary machinery. The boilers are of the single-ended Scotch type, three in number in each vessel, 15 feet 9 inches in diameter by 12 feet 3 inches long between heads, and built for working pressure of 215 pounds of steam per square inch. Each boiler has four 40-inch furnaces of the Morrison suspension type, with separate combustion chambers. There is also a donkey boiler of the Scotch type for harbor use. The furnaces are equipped for burning fuel oil, but there are the necessary fittings to enable the boilers to be converted into coal burners at will. A suction fan gives ventilation to the hold.

The *Kentuckian* and *Georgian* will run between New York and Puerto, Mexico, and the *Honolulan* between San Francisco and Honolulu.—" International Marine Engineering."

The new Southern Pacific liner *El Occidente* was launched by the Newport News Shipbuilding and Drydock Company September 24. This vessel is the last of four to be built for the Southern Pacific Company, *El Sol* and *El Mundo* being in commission and *El Oriente* is now nearing completion. These vessels are 410 by 53 by 36 feet and of 5,700 gross tons.

Largest Ship to Be Dismantled.—The *Shenandoah,* the greatest American wooden ship ever built, or that ever will be built, is lying at Pier 38, Atlantic Dock, New York, 127 days out from San Francisco. She cost $184,000, twenty years ago, but was sold recently for $36,000, to be cut down into a barge to carry coal along the Atlantic Coast. Her twelve thousand square yards of canvas have been unbent for the last time. Her main truck is 192 feet above deck and her main yard is ninety-six feet in length. She is four-masted with six yards on each of her fore, main and mizzen masts, the jigger mast being rigged with fore-and-aft sails. She was

brought here by Capt. James Murphy, who, like the ship, hails from Bath, Me., where she was built by the Sewalls along with the *Roanoke, Rappahannock* and the *Susquehanna.* Two of these were burned and the other foundered at sea. Captain Murphy took her out on her maiden voyage from San Francisco to Havre with 5,164 tons of wheat on board. Five other ships sailed at the same time in the 14,000-mile race, but the *Shenandoah* beat the nearest one by eight days, some of the others being thirty and forty days behind the winner.—"Marine Journal."

The Fore River Shipbuilding Company, Quincy, Mass., launched the tank steamer *Carrier* for the Cuba Distilling Company, September 20. This vessel is 370 by 52 by 22½ feet and is the first to be built in Massachusetts in fifty years for the foreign trade. She will run between Cuban and United States ports.

Transatlantic Steamers with Diesel Engines.—The Hamburg-American Line is building two freight ships equipped with internal-combustion engines, for service between New York and Hamburg. The vessels are of 8,000 tons displacement each, and each will have two 1,500-horsepower engines, giving them a speed of about 12½ knots. This will be the first time that this form of propulsion has been applied to vessels of such size. The engines are being built by the Maschinen Fabrik Augsbourg Company, and are of the Diesel type, using the heavier grades of fuel oil. The oil can be carried in the double bottoms of the ships, doing away with coal bunkers and boiler rooms. There is thus effected great saving in space and weight, and the vessel is made more efficient as a cargo carrier. The engine-room force will be materially reduced. There will be no stokers and few oilers, as the bearings will be lubricated automatically, the oil being filtered and used continuously. It is expected that the vessels will be ready in 1912, and, if successful, will be followed by a passenger ship, propelled by the same method.

OBITUARY.

BEN. WOOD.

If our obituary were under the caption of Benjamin Franklin Wood its designation would probably not be nearly so well recognized as under that of Ben. Wood.

Chief Engineer Ben. Wood was a character, and will long be remembered in the service. He was born in New York in 1836, and learned the trade of machinist at Fletcher & Harrison's in New York City. He was naturally gifted in engineering, and, though never a great student nor a man of scientific pretensions, he was possessed of that quality known in his day as "gumption." He could see into a millstone as far as the wisest man, and, with his rule in his hand, could tell very closely what the diameter of a shaft should be or the amount of draft a key should have, and you might figure on it an hour without being able to prove the error, if it existed.

He was a man of great resource. When the double ender *Sassacus* lost her steering gear in a storm, and the vessel was in danger, "Ben." improvised a temporary rudder which served its purpose. This was not until other improvisions had failed.

During the Civil War "Ben." served on board the *Lancaster* in the Pacific, afterwards on the monitor *Lehigh* and on the double-end gunboats *Sassacus* and *Mohongo*, in the North Atlantic Squadron. After the war had ended he served in the monitor *Dictator;* at the New York Navy Yard; on the flagship *Trenton*; at the Morgan Iron Works, and again at the New York Yard. "Ben" was promoted to be a Chief Engineer in 1883, and did inspection duty at Chester, Penna. After that he served on the old *Kearsarge*; in the European Squadron; on the *Ossipee*; in the N. A. Squadron, and finally at the

Continental Iron Works, Brooklyn, where the so-called torpedo boat *Alarm* was being altered. He was retired in 1892.

Personally "Ben" Wood was a prepossessing man, with very handsome dark eyes, fine features and very dark hair, which never turned gray. He was a reticent man, though rather pugnacious. He was a "stickler" for what was right, but, when beaten, always accepted the result, and was never vindictive.

"Ben" was a model husband, a good son, a good father and a good citizen. He was abstemious to a fault, temperate in his living as in his acts. He was ever regarded as a "good shipmate," and those who had sailed with "Ben" were glad to sail with him again.

Life is short, and in it we are seeking pleasure in our youth and comfort in our old age. Among our shipmates we look for men of capability, integrity and amiability. Ben. Wood possessed the essentials of these, and it is a pity that he could not have lived longer.

G. W. B.

CHIEF ENGINEER WILLIAM B. BROOKS, U. S. N.

The death of Chief Engineer William B. Brooks, U. S. N., which occurred at Erie, Pa., August 23d, was not unexpected. Mr. Brooks had been suffering with a cancer for several years, and his friends knew, as he did, that his days were numbered.

Mr. Brooks was born in Richmond, Va., where he was educated, and where he served an apprenticeship in the Tredegar Iron Works as a machinist. He served a term also in the pattern shop, and was in the draughting room several years. He became an excellent draughtsman, and was on important designs when, in 1852, he was warranted a Third Assistant Engineer in the Navy. Few young men of his day were better qualified to enter the Navy as an Engineer. It must not be forgotten that no books were then to be obtained on mechanical engineering. Every engineer was obliged to make his own text books, and it is not too much to say that the engineers of his day were pioneers.

Mr. Brooks' first duty was on board the *Michigan*, the only U. S. vessel then on the great lakes. He was promoted to Second Assistant Engineer in 1855, and was ordered to the *San Jacinto*, the flag ship of the Asiatic Squadron, which took out the first U. S. Minister to Japan. He was promoted to First Assistant Engineer in 1858, and sent in the Paraguay Expedition, on board the improvised gunboat *New York*, as acting Chief Engineer. The entire expedition was made up of improvised vessels, such as could be purchased for the occasion. Many of them were of doubtful seaworthiness at that time, and would now be condemned on sight.

Mr. Brooks was promoted to Chief Engineer in 1861, and ordered to the *Brooklyn*, one of the formidable vessels of Captain D. G. Farragut's Squadron. Though a Virginian, coming from a Virginia ancestry dating from before the Revolution, he was loyal to the Government and remained faithful to the cause of the Union. He served in all the great battles of the Western Gulf Blockading Squadron, from the capture of New Orleans to the surrender of the fleet in Mobile Bay. It was a time when we had men to fight, and men who put up a gallant fight.

When the war had ended he was sent to the steam sloop *Sacramento* for a cruise in Chinese waters, but that fine vessel was wrecked on the coast of Hindustan and all hands were returned to the United States. He next served in succession as Chief Engineer of the Navy Yard at Portsmouth, N. H.; as Fleet Engineer of the European Squadron, on board the *Franklin* with Farragut; then as Chief Engineer of the Washington Navy Yard; as member of the Engineer Examining Board, and as Fleet Engineer of the North Atlantic Squadron under Rear Admiral Jouett. His last duty previous to retirement was at the Richmond Locomotive Works (successor to the Tredegar Iron Works), where he superintended the building of the engines for the battleship *Texas*, from the designs of a Scotchman. He was retired in 1892.

Mr. Brooks was personally a very handsome man, even in his old age. He was a universal favorite in society, and on

board ship was held in high esteem and much looked up to. He was a quick and ready sketcher with his pencil, and accumulated a great variety of notes and sketches. He had many warm personal friends in official life, among whom was President Hayes. He leaves a widow and one son, Major Brooks, of the U. S. Engineers (Army), whose wife is the daughter of Major General John M. Wilson, late Chief of Engineers, U. S. A. G. W. B.

REAR ADMIRAL CHARLES R. ROELKER.

Rear Admiral Charles R. Roelker died at his home in Washington on the night of September 28th of pneumonia. Thus one more of the men of mark in the old Engineer Corps disappears.

Admiral Roelker was born in Germany in 1841, where he received an academic education. He came to this country immediately after being graduated, and found employment in the draughting room of the De Lamater Iron Works, in New York, where his older brother was superintendent. He was a Greek and Latin scholar, and had no difficulty in becoming proficient in English.

He entered the Navy as a Third Assistant Engineer December 27, 1862, and was at once ordered to the *Susquehanna*. He saw further service during the Civil War on board the *Sonoma* and *Nipsic* and in the draughting room of the Bureau of Steam Engineering. He was promoted to Second Assistant Engineer in 1864, and to First Assistant in 1868. He served on board the *Tallapoosa*, *Frolic*, *Fortune*, *Shenandoah*, *Franklin* and *Congress*, and again in the Bureau of Steam Engineering.

In 1882 he was granted three months leave of absence, which period he spent in Europe, mostly in Germany. He always said he was an American by preference, not by accident, as were those of us who were born here, but after his

short sojourn in his native land, where he found so few of his old friends, and those who were left had grown away from him and his interests, returned to the United States with renewed enthusiasm for his chosen country.

When the Advisory Board was created, in 1882, Mr. Roelker was detailed in charge of the draughting room of that Board, where he was responsible for the designs. This duty was continued until 1887, when the Board's work was finished, and he was ordered to the deep-sea exploring ship *Albatross*. He was promoted to the grade of Chief Engineer (rank of Lieutenant Commander) in 1890, and was ordered to superintendent of construction of the machinery for the cruiser *Raleigh*, on the completion of which duty he served as chief engineer of that vessel for three years. His next duty was that of a member of the Board of Inspection and Survey.

He was promoted to the rank of Commander (in the Engineer Corps) in 1899, and, later, his rank was changed, by law, to that of commander in the line of the Navy. He became Captain in 1902, and, on the 23d of September, 1903, was retired with the rank of Rear Admiral.

Mr. Roelker continued on duty after his retirement, under the Attorney General and under the Bureau of Fisheries. Later he served as senior member of a Board in connection with the revision of general rules and regulations prescribed by the Board of Supervising Inspectors of the Steamboat Inspection Service.

The subject of this sketch was one of the finest draughtsmen in the corps, and could use the brush with the same facility as the drawing pen. He also made the distinctive design (propeller and foul anchor) used by the Bureau of Steam Engineering; and when the corps was attacked by certain newspapers he suggested many effective points in its defense. He was a member of the American Society of Naval Engineers and contributed several valuable papers for our journal.

He was popular wherever he served, made warm friends in

every ship, and was a modest, diffident man, reticent and sensitive.

A musician of a high order, he could play difficult music on the piano at sight and with a touch that was charming.

His duty on the Board of Inspection, which took him on many trial trips, was severe on a man of his age and size, for as he grew older he became rather stout; his duties on trial trips kept him much in the engine room, which was always hot, and the change to a cool atmosphere several times gave him severe colds which came near resulting fatally. He was an uncomplaining man, so that few ever knew of his sufferings.

Mr. Roelker's home life was beautiful. When his children were small his Sundays were spent with them in the woods or at the seashore, where it was his pleasure to teach them to collect and to be interested in plants, butterflies, etc., and to talk about things, not about people. He married the daughter of Dr. Porter, a physician in Georgetown, who survives him with three sons and three daughters.

Mr. Roelker had a remarkably retentive memory. In his early days, when the literature on mechanical engineering was very small, it was the custom for engineers to make their own books and formulae, and fill sketch books with drawings of anything at all likely to be useful to them. While the rest of us were saving our sketches Mr. Roelker simply remembered his and could reproduce from memory what he needed.

Though apparently phlegmatic he was really a nervous man. He seemed so eager to conceal his infirmities that, at times, he appeared apathetic. G. W. B.

FREDERICK MERIAM WHEELER.

Frederick Meriam Wheeler died September 15th at his summer home, Westhampton, Long Island, of heart disease, in his sixty-second year. Mr. Wheeler was born in Brooklyn. His father, John Wheeler, a native of England, married Jane, eldest daughter of Francis W. Meriam, of Brooklyn, a direct

descendant of Joseph Meriam, who came to America on the good ship *Castle* in 1638, and settled in Concord, Mass.

Mr. Wheeler was graduated at Summit Academy, N. J., and subsequently attended the Polytechnic Institute, of Brooklyn. He studied mechanical engineering for four years under Henry J. Davison, of New York City—one of the most noted mechanical engineers of this country. He afterward took up hydraulic and marine engineering as a specialty, and for over thirty-four years was associated with the George F. Blake Manufacturing Company. He was a director and secretary of the Company, and later a director of the International Steam Pump Works, which corporation absorbed the Blake Company, the Worthington and other hydraulic works. Mr. Wheeler was the inventor of the Wheeler patent surface condenser, which is largely in use throughout this country and abroad, and is in extensive use by the United States Navy. It is also used for testing and laboratory work in nearly all the mechanical institutions and schools of technology throughout this country and Europe. Mr. Wheeler organized the Wheeler Condenser and Engineering Company, whose works are located at Carteret, N. J. Mr. Wheeler has also been officially connected with the Ludlow Valve Manufacturing Company. He was one of the charter members of the American Society of Mechanical Engineers and the Society of Naval Architects and Marine Engineers. He was a member of the Engineers' Club and was for many years an Associate Member of the American Society of Naval Engineers, taking an active interest in the affairs of the Society. He served as a member of the Advisory Council of the Engineering Congress at the Columbian Exposition. After leaving Brooklyn, his native city, Mr. Wheeler came to Montclair, and married, in 1876, Florence, the eldest daughter of the late Charles K. Willmer.

Mr. Wheeler was prominent in social circles in Montclair for many years. He was the founder of the Montclair Equestrian Club, which subsequently became the Montclair and Essex County Country Club, in Orange. He was also one of

the founders of the Outlook Club, and one of the early members of the Montclair Club, and for many years on its board of directors. While assisting in the various public improvements from time to time, the subject which interested Mr. Wheeler most was that of public parks for Montclair, which subject he agitated for many years until it became a reality. In other public matters he was also heartily enthusiastic, and had always worked in the interests of good government. In church matters he served on the vestry of St. Luke's Episcopal Church for many years, and was a member of the original building committee of the present church edifice.

Mr. Wheeler is survived by Mrs. Wheeler, two daughters and two sons. The daughters are Beatrice Molineux, now Mrs. L. H. Spier, and Gladys Willmer. The sons are Knight Willmer and Cottrell Corbett Wheeler.

CONTRIBUTED.

BOOK NOTICE.

PRINCIPLES OF WIRELESS TELEGRAPHY.—By GEORGE W. PIERCE, A. M., Ph. D. McGRAW-HILL BOOK CO., New York.

This is an excellent practical treatise of 328 pages, in which the history and development of the art is treated in detail. The descriptions are scientific enough to convey a clear idea of the subjects treated, but they are not too technical, nor the mathematics too involved, to be understood by the average student. The book is well illustrated, and is recommended to readers of the JOURNAL who desire to study the underlying principles of wireless telegraphy, without having to hunt through a maze of technicalities.

ASSOCIATION NOTES.

On Tuesday, October 4, 1910, a regular meeting was held at the Navy Department, Washington, D. C., at which nominations were made for officers of the Society for the coming year, as follows:

For President:

Engineer-in-Chief H. I. CONE, U. S. Navy.

For Secretary-Treasurer:

Lieutenant Commander U. T. HOLMES, U. S. Navy.
Lieutenant Commander JOHN HALLIGAN, U. S. Navy.

For Members of Council:

Rear Admiral GEORGE W. BAIRD, U. S. Navy, Retired.
Engineer-in-Chief C. A. MCALLISTER, U. S. R. C. S.
Captain R. S. GRIFFIN, U. S. Navy.
Commander C. W. DYSON, U. S. Navy.
Commander W. STROTHER SMITH, U. S. Navy.
Commander EMIL THEISS, U. S. Navy.
Commander W. W. WHITE, U. S. Navy, Retired.

A voting slip has been mailed to each member, and, as stated therein, the votes will be opened and counted at the Annual Meeting of the Society, to be held at the Navy Department, Washington, D. C., at 4:30 P. M., Friday, December 30, 1910.